ASHRAE Student Design Competition 2015
HVAC System Selection

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University of Nebraska

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1.0: Executive Summary

The purpose of this report is to analyze, compare, and select the most appropriate heating, ventilation, and air conditioning (HVAC) system for a new three-story junior college located in Doha, Qatar. This facility houses classrooms, administration spaces, faculty offices, special instruction areas, and information technology areas. The building consists of approximately 48,000 square feet distributed on three levels, including woodworking/carpentry shop and welding shop spaces on the first level.

The challenge of this system selection process includes fulfilling the requirements of the Owner's Project Requirements (OPR) while meeting ASHRAE Standards. The Owner requires that the system provide a sustainable design, taking into account energy efficiency, indoor environmental quality and safety, occupant comfort, functionality, longevity, flexibility, serviceability, and maintainability. The selected system shall have the lowest life cycle cost among the system options. Additionally, the owner requests that the energy consumption of the building is reduced at least 15% more than required by ASHRAE Standard 90.1.

Load calculations and an energy simulation were completed using Trane TRACE 700. To determine the best HVAC system for the building, three different systems were considered. The first simulation in TRACE was performed on the ASHRAE Standard 90.1 Baseline system of a packaged variable air volume DX cooling rooftop unit, with variable air volume hot water reheat terminal units. Three other systems that fulfill the owner's requirements were analyzed and compared to both the Baseline and each other. These systems are:

Option 1 – VAV air handling system with electric reheat and a chiller utilizing thermal ice storage with a water fountain spray cooling tower

Option 2 – Fan coil units with dedicated outside air system and a chiller utilizing thermal ice storage with a water fountain spray cooling tower

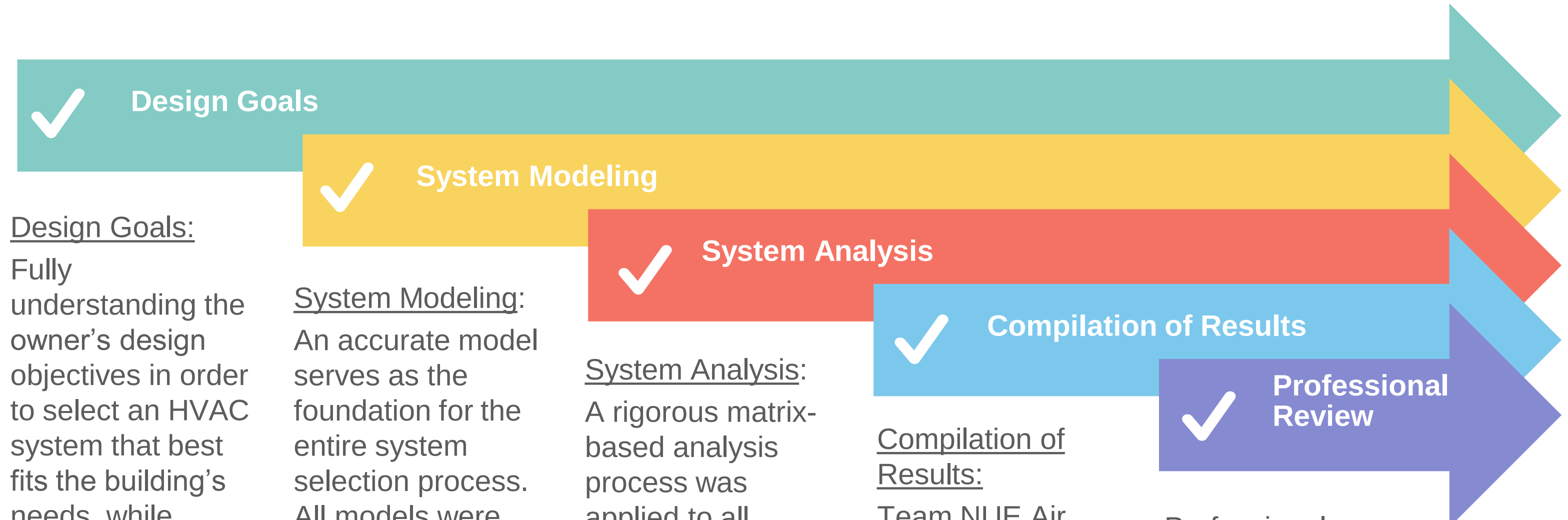
Option 3 – Water source heat pump system with dedicated outside air system, utilizing a closed seawater loop field

To compare the systems, a decision matrix was created including categories of initial cost, life cycle cost, temperature control, humidity control, controllability, indoor air quality, system reliability, design flexibility, spatial requirement, noise, sustainability, high performance building compliance, maintenance requirement, and synergy with architecture. The HVAC system with the highest point value was selected.

Based on this analysis, Option 3 - the water source heat pump system with dedicated outside air system, utilizing a closed seawater loop field - is the most appropriate HVAC system for the new junior college in Doha, Qatar. This system showed a 44% energy improvement over the Baseline. This system is projected to cost \$3.8 million over a 50 year period, which is \$695,000 less than Option 1 - VAV air handling system with thermal ice storage - and \$220,600 less than Option 2 - FCU with thermal ice storage.

Team NUE Air's design process flows as follows: establish design goals, model three system options, analyze the systems, compile appropriate results, and review the results with industry professionals. This process is summarized below. Each section of this report is summarized below.

Section 2.0: Introduction identifies the building, the owner's requirements, and the ASHRAE standards that require compliance.
Section 3.0: Design Parameters introduces the conditions to which each system option is designed.
Section 4.0: Design Goals outlines Team NUE Air's goals in designing each system option. This section also specifies and explains elements of the design that comply with these goals.
Section 5.0: Building Design defines elements of the building that directly affect the design of the building's HVAC system.
Section 6.0: Load Calculations describes Team NUE Air's method for modeling the building in Trace TRACE 700.
Section 7.0: Major System Components details major components that are combined in various manners to compose each system option.
Section 8.0: Systems Considered establishes Team NUE Air's three system options.
Section 9.0: Cost outlines the life cycle cost analysis for each of the three system options.
Section 10.0: Decision Matrix describes Team NUE Air's detailed decision-making process to select the optimal HVAC system.
Section 11.0: Selected System identifies the system that Team NUE Air chose as the optimal system.
Section 12.0: Conclusion recommends the optimal system for the building.



Design Goals: Fully understanding the owner's design objectives in order to select an HVAC system that best fits the building's needs, while maintaining efficient and unique design.

System Modeling: An accurate model serves as the foundation for the entire system selection process. All models were reviewed thoroughly to ensure accuracy in this critical section.

System Analysis: A rigorous matrix-based analysis process was applied to all systems. System function, efficiency, and environmental impact were a few of the many categories used to compare the different systems.

Compilation of Results: Team NUE Air went through many revisions to ensure a clear, concise, and organized deliverable was produced to convey the results to the owner.

Professional Review: The submission of an accurate product is paramount within the engineering industry. The validity of the deliverable was ensured by professional industry review.

2.0: Introduction

2.1: Building Description

The new educational facility is a three-story junior college located in Doha, Qatar. The building has a floor area of approximately 48,000 square feet. The first floor is approximately 18,000 square feet and includes classrooms, offices, conference rooms, a library/media center, a computer lab, and a carpentry/woodworking/welding shop. The second floor is approximately 15,000 square feet and includes classrooms, a conference room, and a student gathering area. Similarly, the third floor is approximately 15,000 square feet and includes classrooms, a conference room, a student gathering area, and as a computer lab. A location within Doha near the Persian Gulf was assumed as discussed in the design competition website's FAQ page. See Figure 2.1 for a rendering of the building.

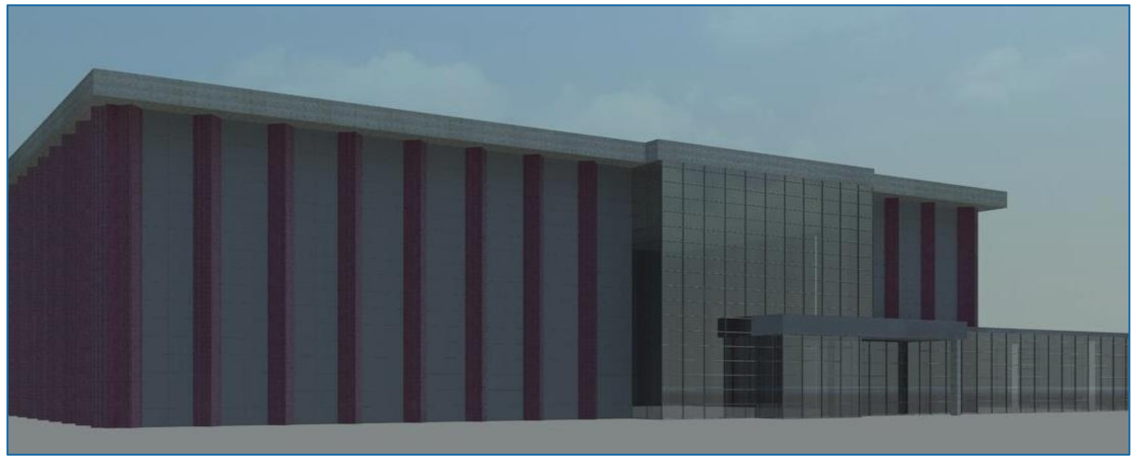


Figure 2.1: Junior College in Doha, Qatar
(source: authors)

2.2: Owner's Requirements

Team NUE Air's major design considerations were based on the Owner's Project Requirements. These criteria set a baseline for the functionality of the HVAC system within the building. The Owner's Project Requirements include:

1) Sustainable design through energy efficiency, indoor environmental quality and safety, occupant comfort, functionality, longevity, flexibility, and serviceability/maintainability
2) Reduce energy consumption by at least 15% more than the baseline prescribed in ASHRAE Standard 90.1
3) Lowest possible life cycle cost, considering initial cost of materials and operation and maintenance costs
4) Operation and maintenance requirements with easy serviceability
5) Indoor environmental quality that facilitates occupant productivity, comfort, and safety
6) Noise control encompassing superior acoustic criteria in classrooms and minimal sound transmission from shop areas
7) Photovoltaic array that meets 5% of the building's total energy needs
8) Compliance with **ASHRAE Standard 55-2004**, **Standard 62.1-2010**, **Standard 90.1-2010**, and **Standard 189.1-2011**

2.3: Reference Standards

2.3.1: ASHRAE Standard 55

ASHRAE Standard 55 was used to determine the proper thermal environment for occupant comfort. The factors that define thermal comfort include: metabolic rate, air temperature, air speed, humidity, clothing insulation, and radiant temperature. Metabolic rates were assumed to be between 1.0 met and 1.3 met and clothing insulation values were assumed to be between 0.5 clo and 1.0 clo. The following formulas from Standard 55 Section 5.2.1.1 were used to find the acceptable operative temperature range:

$$T_{max,Icl} = \frac{(Icl - 0.5clo) * T_{max,1.0clo} + (1.0clo - Icl) * T_{max0.5clo}}{0.5clo}$$

$$T_{min,Icl} = \frac{(Icl - 0.5clo) * T_{min,1.0clo} + (1.0clo - Icl) * T_{min0.5clo}}{0.5clo}$$

Building compliance with Standard 55 is described in Section 4.3.

2.3.2: ASHRAE Standard 62.1

ASHRAE Standard 62.1 was used to determine minimum ventilation and exhaust rates for acceptable indoor air quality to minimize adverse occupant health effects. Calculations for the minimum outdoor air requirement were completed. Standard 62.1 Section 6.2 presents the following formula, with values found in Table 6.1 and Table 6.2 of ASHRAE Standard 62.1:

$$V_{oz} = \frac{R_p P_z + R_a A_z}{E_z}$$

Compliance with this Standard was achieved by supplying each room with the volume of ventilated air specified by the above equation for the options that included a DOAS. For Option 1, multiple space effects were considered by applying the equations in Section 6.2.5 of the standard. Exhaust rates for areas with high pollutant generation rates were given by a similar manner via ASHRAE Standard 62.1 Table 6.4, except for the paint room, where other specific requirements dictated exhaust airflow. Here an exhaust rate of 10 L/sec/person was used, and all ventilation air passes through a standalone HVAC unit as discussed in Section 6.3 of this document. Externally, all ventilation air inlets and outlets are separated as per ASHRAE Standard 62.1 Table 5-1, taking care to keep Class 4 air (as seen in the paint booth) outlets 30' from any inlet.

For transfer air, flow rates are based upon air class. ASHRAE Standard 62.1 Section 5.16 states that it is acceptable to utilize transfer air in the design at locations specified in Section 6.3 of this document. This is possible because Class 1 corridor air is able to be moved from space to space. ASHRAE Standard 62.1 Section 5.16 also states that Class 2 restroom air cannot be transferred to the adjacent corridor. As Class 4 air cannot be transferred to any space, the design decision to fully isolate and ventilate the paint booth is also supported.

2.3.3: ASHRAE Standard 90.1

ASHRAE Standard 90.1 was used to specify a Baseline system to which the three system options were compared. New buildings per ASHRAE 90.1 Section 4.2 must comply with sections 5, 6, 7, 8, 9, and 10 or 11. Of these, Sections 5, 6, and 9 fall within the mechanical design. Compliance of these sections was executed by:

1) Section 5 - U-value hand calculations (see Appendix D.3)
2) Section 6 - Digital controls with automatic shutdown for each zone, motorized dampers that automatically close and are tied into the fire and smoke detection systems, demand control ventilation (DCV) will be used for areas more than 500 ft^2 with higher than 40 people per 1000 ft^2 , and variable speed drives or two-speed motors for AHUs and FCUs
3) Section 9 - Lighting is controlled with scheduled lights and occupancy sensors in most rooms.

2.3.4: ASHRAE Standard 189.1

ASHRAE Standard 189.1 was used to determine the minimum equipment efficiencies. Standard 189.1 aligns with some LEED requirements. Team NUE Air's implementation of LEED (see Section 4.10) aided in mechanical design compliance with Standard 189.1.

2.3.5: ASHRAE GreenGuide

ASHRAE defines a green building as one that achieves high performance over the full life cycle in the following areas: minimal consumption of nonrenewable and depletable natural resources, minimal atmospheric emissions, minimal discharge of harmful wastes including those from demolition of the building, and minimal negative impacts on site ecosystems. Team NUE Air used the ASHRAE GreenGuide to design a building as near to the green building definition as possible. The ASHRAE GreenGuide outlines a design process that team NUE Air utilized in the following categories:

1) Commissioning - executed through the implementation of LEED
2) High Performance Energy Strategies - executed through the analysis of three energy-efficient HVAC systems
3) Occupant Comfort and Health - executed through compliance with ASHRAE Standard 55 in combination with "green tips" from the Guide, such as energy recovery and DOAS
4) Integrated Building Design and Energy Sources – executed through the use of a photovoltaic (PV) louver system

3.0: Design Parameters

3.1: Weather Data

Doha, Qatar has a hot desert climate, deemed equivalent to Climate Zone 1B in the United States. Doha's warm season lasts from early May to late September with an average daily high temperature of approximately 99°F. The hottest day of the year usually occurs in early July with an average high of 106°F and low of 87°F. Doha's cold season lasts from early December to early March with an average daily high temperature of approximately 77°F. The coldest day of the year usually occurs in late January with an average low of 57°F and high of 70°F. Figure 3.1 shows the average daily high and low temperatures throughout the year in Doha

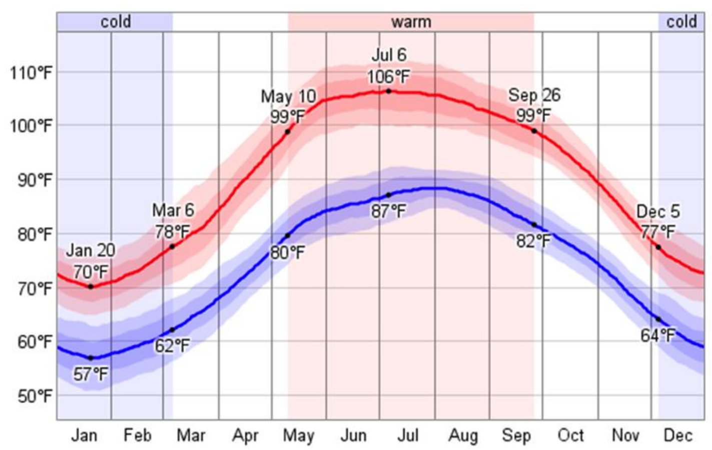


Figure 3.1: Daily High and Low Temperatures
(source: Weather Spark)

ASHRAE Standard 90.1 Section 5.1.4.2 defines how to determine the climate zone number and letter. While ASHRAE documents do not directly list Doha's climate zone, they do provide the data necessary to determine the city's ASHRAE-defined zone. Table B-4: International Climate Zone Definitions provides thermal criteria for each international climate zone. Analysis of this figure shows the need for cooling degree day base 50 data (CDD_{50}), which is the annual amount of degrees per day that the mean temperature is over 50°F. Examination of Doha's climatic data and subsequent calculations (see Appendix D.1) show that Doha is within the Very Hot-Dry (1B) climate zone. This definition implies that Doha buildings are cooling dominated.

3.2: Outdoor Design Conditions

ASHRAE Handbook of Fundamentals 2013 provides design conditions for the Hamad International Airport in Doha. Cooling design conditions were based on the 2% criteria, as specified by the OPR. Heating design conditions were not specified and have been based on the 99.6% design condition. The conservative heating design condition was based on the low first cost associated with adding heating capacity. The cooling and heating design conditions are shown in Table 3.1. These design conditions were used in the load calculations to size the HVAC equipment.

Table 3.1: Cooling and Heating Design Conditions

Cooling: 2%		Heating: 99.6%
Dry Bulb (°F)	Mean Coincident Wet Bulb (°F)	Dry Bulb (°F)
106.1	72.9	52.0

(source: 2009 ASHRAE Handbook of Fundamentals Supplemental CDROM)

3.3: Indoor Design Conditions

The indoor design conditions for the building were provided in the OPR and can be seen in Table 3.2. These conditions were used within the TRACE model for load calculations and the energy model. Thermostat schedules for heat and cooling were designed around these conditions.

Table 3.2: Indoor Design Conditions

Area	Summer	Winter
Office & Administrative Support Spaces	73.4°F (23°C) DB 50% RH	70.0°F (21°C) DB
Classroom & Study Spaces	73.4°F (23°C) DB 50% RH	70.0°F (21°C) DB
Library	73.4°F (23°C) DB 50% RH	70.0°F (21°C) DB
Special Instruction Spaces	78.8°F (26°C) DB 55% RH	73.4°F (23°C) DB
Information Technology Support Spaces	73.4°F (23°C) DB 50% RH	73.4°F (23°C) DB 50% RH

(source: Owner's Project Requirements)

3.4: Ventilation Requirements

Outdoor air and exhaust ventilation rates were based on ASHRAE Standard 62.1 Sections 6.1 and 6.2. Table 6-1 gives the Minimum Ventilation Rates in Breathing Zones and Section 6.2.2.1 outlines the formulae and procedure for calculating the outdoor airflow required in the breathing zone. ASHRAE Standard 62.1 Equation 6-1:

$$V_{bz}=R_p*P_z+R_a*A_z$$

where R_p is the outdoor airflow rate required per person and R_a is the outdoor airflow rate required per unit area given in ASHRAE 62.1 Table 6-1 for each zone's occupancy category and room type.

3.5: Hours of Operation

The hours of operation of the building were used in creating an accurate energy model. The occupancy hours can be seen in Table 3.3.

Table 3.3: Hours of Operation

Area	Occupancy hours
Office & Administrative Support Spaces	Monday - Friday: 7 am - 6 pm Saturday: 8 am - 1 pm
Classroom & Study Spaces	Monday - Friday: 8 am - 5 pm
Library	Monday - Friday: 8 am - 5 pm
Special Instruction Spaces	Monday - Friday: 8 am - 3 pm

(source: Owner's Project Requirements)

Utilization schedules in TRACE were defined to account for occupancy, lighting, ventilation, infiltration, and miscellaneous loads based on the hours of operation of the building. Per ASHRAE Standard 90.1 Section 6.4.3.3, off-hour controls with automatic shutdown, setback controls, optimum start controls, and zone isolation are incorporated to override controls that are based on these schedules.

3.6: Utility Rates

The OPR provided the cost for electricity, natural gas, city water, and city sewer. These are shown in Table 3.4.

Table 3.4: Utility Rates

Type	Rate	Annual Increase
On-Peak Electricity Consumption	$0.1614/kWh	3.5%
On-Peak Electricity Demand	$9.75/kW	3.5%
Off-Peak Electricity Consumption	$0.085/kWh	3.5%
Natural Gas	$7.91/Mcf	3%
City Water	$0.02/ft³	2.5%
City Sewer	$0.003/gal	2.5%

(source: Owner's Project Requirements)

On-peak time period for electricity is from 9:00 a.m. to 7:00 p.m. Monday through Saturday. Off-peak time period for electricity includes all other times and does not have a demand charge. Per the OPR, the utility escalation rate was based on a 10 year average increase for utility provided in the New York City area.

A thermal storage schedule was created for the two system options that utilize thermal storage. The system charges from 7 p.m. to 9 a.m. and satisfies load from 9 a.m. to 7 p.m.

4.0: Design Goals

Team NUE Air created a list of design goals to ensure that all aspects of the project were considered and properly analyzed.

4.1: Low Life Cycle Cost

According to the OPR, the building's allocated total budget was $200 per square foot. The life cycle cost analysis was completed with a building life of 50 years, a 7% return on investment, and a 3% inflation rate. First costs were calculated from prices provided by industry professionals, Engineering News-Record (ENR) Mechanical/Electrical Square Foot Costbook 2015 Edition, and direct quotes from manufacturers for the large mechanical components. The operation, maintenance, and replacement costs of each system were compared over 50 years to comply with ASHRAE Standard 189.1 for a "Long Life" building. A 50-year life cycle cost of each system was also calculated and compared. A return on investment chart based on present worth was created to show the year that the recommended system life cycle cost payback would exceed the other systems (see Section 9.0).

Present worth analysis was used for all costs. Simple present worth cost is defined as the equation: $P=(1+i)^{-n}$, where i is the interest rate and n is the number of years. The present worth for each system was calculated using the energy usage costs per year provided in TRACE for each system and the utility rates provided in the OPR. See Section 9.0 for the full life cycle cost results of each system option.

4.2: Low Environmental Impact

The environmental impact of each system option was considered for selection of refrigerant type, water usage, total building energy consumption, and lighting controls.

Environmental effects such as ozone depletion were taken into consideration when selecting a refrigerant. While HCFC R-22 is more efficient than hydro fluorocarbon (HFC) refrigerants, R-22 was not chosen due to its 2010 forbiddance of use in new HVAC equipment in compliance with the EPAs phase-out schedule under Title VI of the Clean Air Act. As a secondary choice HFC refrigerants R-410a and R-134a were chosen for the system, neither of which contribute to ozone depletion.

The chosen water source heat pump system has a clear advantage in terms of water usage. Despite transferring heat via water, the entire system is closed-loop and does not contaminate the environment. Additionally, this system eliminates the use of evaporative cooling. In this way, savings are made in water costs, which is imperative for cities such as Doha, which lack naturally occurring fresh water.

Team NUE Air focused on minimizing total building energy consumption in order to minimize carbon footprint. The total building energy consumption for all three system options was found using TRACE. These values are summarized in Table 4.1. Each system option produced an energy improvement over the Baseline, thus reducing overall environmental impact.

Table 4.1: Total Building Energy

System	Total Building Energy (kBtu/yr)	Improvement Over Baseline	Building Energy per Square Foot (kBtu/yr)/s.f.
Baseline	2,557,945	N/A	55.09
VAV AHU with Thermal Storage	2,138,311	17%	46.05
FCU with Thermal Storage	1,830,675	28%	39.43
WSHP with Sea Loop	1,445,622	44%	31.14

(source: authors)

Lighting controls for all spaces were designed in compliance with ASHRAE Standard 90.1 Section 9.4 where each space will have a manual on-switch that is readily accessible to occupants. An occupancy sensor is present in each room with the exception of the shop room and mechanical/electrical rooms, which will be time-of-day controlled with a temporary one-hour override. This override will be located at the thermostat that switches HVAC and lighting systems into occupied mode. Manual switches in corridors, electrical/mechanical rooms, and stairways will have an on and off setting, and all other rooms will have an additional third setting of 50% power that turns half of the lights off. Occupancy sensors automatically turn lights off within 30 minutes of all occupants leaving the space, and daylight harvesting shuts off lights adjacent to windows if the lighting level is high enough.

4.3: Comfort and Indoor Environmental Quality

The comfort and indoor environmental quality of the classrooms and office spaces will differ from the woodworking and metal shops, due to the function of the different spaces. The classrooms and offices require a higher level of control and reduced noise levels, whereas the shop area will be designed to reduce fumes and fugitive dust generated by the equipment.

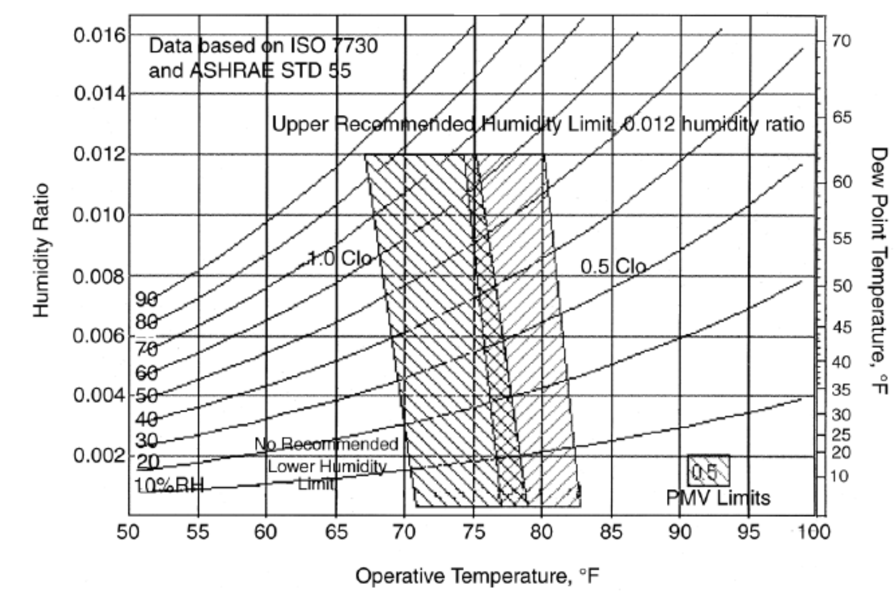


Figure 4.1: Acceptable range of operative temperature and humidity (source: ASHRAE Standard 55-2004)

Table 3.2 shows that conditions fall within the acceptable range of operative temperatures according to ASHRAE Standard 55 Section 5.2.1.1. The temperature range shown in Figure 4.1 is for 80% occupant acceptability and comfort with humidity levels between 40% and 60%. The system options were designed to meet the given temperature and relative humidity requirements that comply with Standard 55. Figure 4.1 shows the acceptable range of operative temperature and humidity for this type of space.

4.0: Design Goals

Section 6 of Standard 55 requires complete plans, equipment data, descriptions, and operation and maintenance data for the building systems to be provided and maintained. This will happen at the completion of the design phase. Also, evaluations of the thermal environment will take place after the construction of the building, to confirm that the mechanical system is operating properly to provide the most comfortable environment for the occupants. They include measurements of temperature and humidity in zones when the zone is loaded to at least 50% of the design load and heat gain by occupants is simulated.

4.4: Creative High Performance Green Design

PV Louvers – Applicable to Options 1, 2, & 3

To meet the criterion that a minimum of 5% of the building's total electrical load provided by photovoltaic (PV) panels, a louver-based PV system is applied to the south facade of the structure. This south-facing design will maximize the system's exposure to solar radiation. Additionally, the louver-based system provides shading to the areas most vulnerable to solar heat gain: south-facing windows. Analysis using the clean energy management software RETScreen shows an expected total energy output of 53,700 kWh.

Water Fountain Spray Cooling Tower – Applicable to Options 1 & 2

As indicated in Section 3.1, Doha is very hot and dry, which allows for the consideration of a water fountain spray cooling tower. (It should be noted that an ASHRAE defined dry climate does not necessarily mean a climate with low humidity, but data show that the humidity levels in Doha are still low enough to justify this design.) This creative design functions by utilizing evaporative cooling from architecturally-pleasing water features to provide adiabatic cooling for condenser water to supplement the cooling tower. Evaporative cooling occurs partially via the large, relatively still water pond, and more prominently (due to spray and higher velocity) via the actual water jet. Calculations in Appendix D.2 show that when combined, this water fountain spray cooling tower could provide 30.6 tons of additional cooling to the structure. A heat exchanger would be used to separate the water fountain spray cooling tower from the cooling tower supply water, thus preventing contamination.

Closed Seawater Loop field – Applicable to Option 3

Doha's location adjacent to the Persian Gulf allows for another less common design choice through the use of a closed seawater loop field that rejects heat from a heat pump loop. As Doha sees nearly year-round cooling loads, a traditional geothermal loop would continuously reject heat to the soil. Without a substantial heating season to offset the heat gain of the soil, heat soak would occur, thus decreasing the efficiency of the loop.

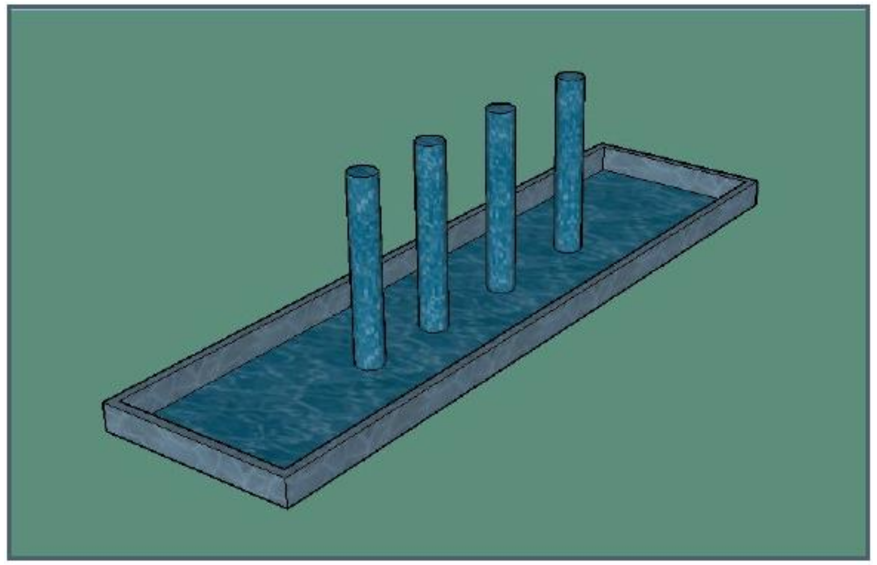


Figure 4.2: Water Fountain Spray Cooling Tower

The Persian Gulf maintains a constant daily temperature and, when combined with water currents, eliminates heat soak. One concern with this technology is high condenser water temperature. Figure 4.3 shows the average surface sea water temperatures, which peaks at 95°F. Fortunately, the water's peak temperature lags behind the air peak by nearly two months. This means that there is a larger ΔT between the system and the water temperature, which raises the system efficiency.

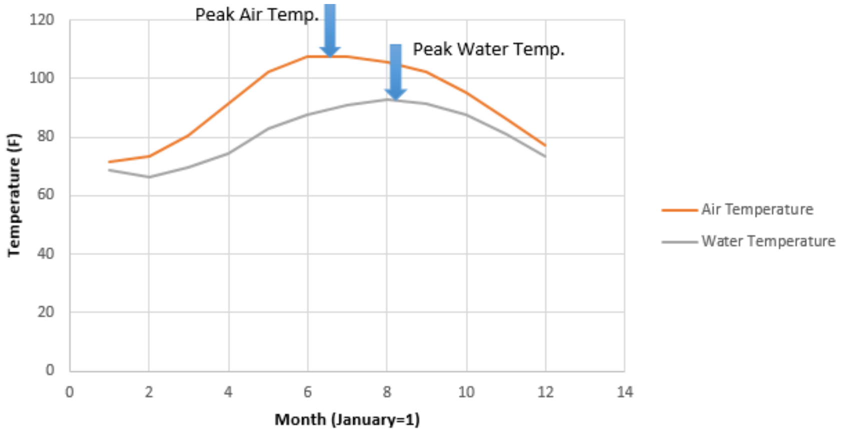


Figure 4.3: Lag of Water Temperature (source: authors)

4.5: Synergy with Architecture

PV Louvers – Applicable to Options 1, 2, & 3

The PV louver system provides an architecturally-pleasing incorporation of PV panels. Traditional systems require large panels to be placed near or on the structure, which can detract from the appearance of the building. Team NUE Air understood that shading would be required on the large glazed areas of the southern facade, making the addition of the PV louver system an obvious choice. With the louvers being a single unit with the PV system, a cohesive, sleek design is created. Additionally, the costs of a standalone shading system is eliminated. Figure 4.4 outlines the areas of the south facade that will contain the PV louver system. Figure 4.5 shows a representation of what the louver system would look like on an exterior panel of the building.

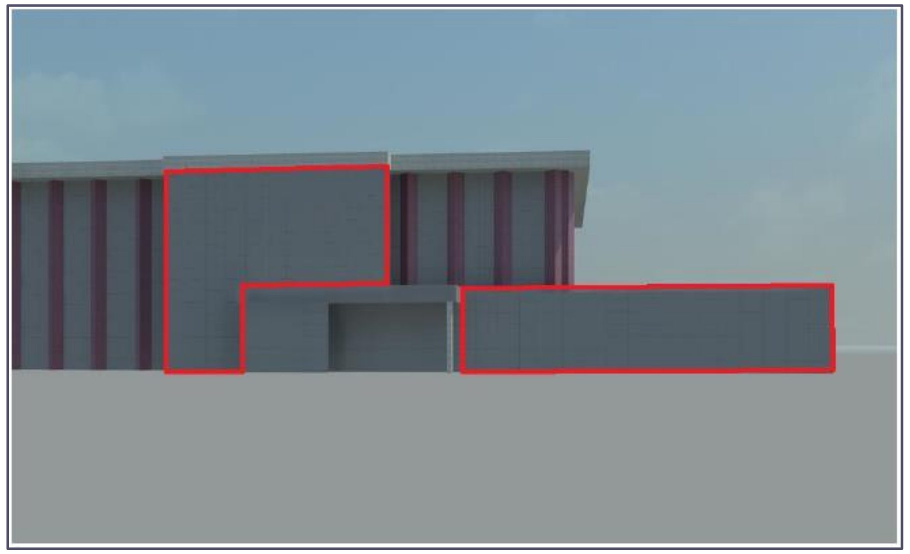


Figure 4.4: PV Louver Location

Water Fountain Spray Cooling Tower – Applicable to Options 1 & 2

The water fountain spray cooling tower introduced in Section 4.4 is another dual purpose creative green design that has both performance and architectural benefits. The very nature of the system - involving rising, spraying jets of water - serves to welcome occupants into the building with a pleasant visual experience. By providing initial cooling, the maximum load on the cooling tower is decreased, thus allowing for a smaller size of cooling tower to be installed. The water fountain spray cooling tower provides a beautiful method of evaporative cooling and decreases both the overall visual and cost impact of the cooling tower through the reduction in size. Such a water fountain spray cooling tower has been implemented at the Museum of Islamic Art in Doha, Qatar (see photo in Figure 4.6).

4.0: Design Goals

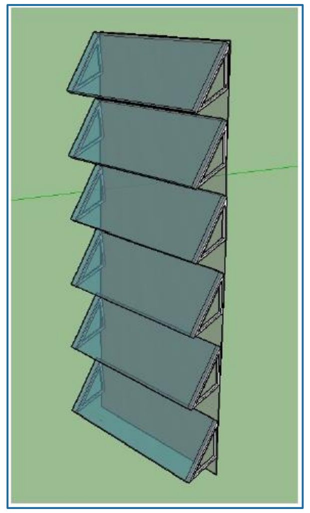


Figure 4.5: PV Louver Representation
(source: authors)

Figure 4.6: Water fountain spray cooling tower at the Museum of Islamic Art in Doha, Qatar
(source: Industry Advisor/ASHRAE Mentor Joe T. Hazel)

Closed Seawater Loop field – Applicable to Option 3
A seawater heat pump system has the advantage of not having a cooling tower to incorporate into the architecture. Unlike a system utilizing a large primary cooling tower for heat rejection, a heat pump system can reject its heat to a large reservoir of moving water (or in this case, the Persian Gulf). The heat pump system will have a small dedicated outdoor air system (DOAS) on the roof to treat incoming outdoor air before it reaches the heat pumps. Accompanying the DOAS unit will be an enthalpy wheel type of energy recovery ventilator detailed in Section 7.1 to dramatically increase system efficiency. Since the DOAS unit is simply supplementary to the heat pumps, it is small in size, thus allowing it to be placed centrally on the roof with minimal structural impact. Due to the building's height, the DOAS will be completely out of sight at this location.

Team NUE Air considered the internal architectural impacts of each system in addition to the external architecture. The building space usage and consistency of occupant comfort are addressed in the form of two major categories:

1) Minimize use of occupied space for mechanical systems: Careful consideration of system type and component location is vital to avoid the need to use preferred occupied space for mechanical space, as this is both expensive and visually unpleasant. With the selection and zoning of the water source heat pump system, team NUE Air was able to eliminate external cooling towers while keeping internal heat pumps small in size, allowing for more flexibility in their placement. The larger, harder to service and conceal heat pumps (serving locations such as the library and construction rooms) are placed centrally in the mechanical rooms, while the smaller units serving individual rooms will be placed in the corridors adjacent to their rooms, eliminating long duct runs and allowing the units to be serviced outside of the room, avoiding the distractions of a closet-mounted unit.

2) Minimize impact of diffusers: Diffusers are essential for the distribution of treated air in mechanical systems, and must be carefully considered in their placement in order to maintain an unobtrusive environment for building occupants. Typical design practice was used which considers diffuser spacing in order to avoid airstream collisions, and coordination with lighting will be considered to create a symmetric design. Finally, the use of transfer air in the construction rooms allows for fewer supply diffusers needed in the space, giving a more seamless look to the area.

4.6: High Efficiency

High efficiency was considered to ensure selection of an optimally-performing system. Ambient air temperature (both dry bulb and wet bulb) was considered when selecting system components. The amount of cooling provided by an air-cooled condenser is a function of the ambient dry bulb temperature, while the amount of cooling provided by a water-cooled condenser is determined by the ambient wet-bulb temperature. A water-cooled condenser is more efficient than an air-cooled condenser due to the low monthly mean wet bulb temperature, which contributes to a larger ΔT in the heat transfer equation: $Q=mC_p(\Delta T)$. Thus, a cooling tower was chosen for System Options 1 and 2.

4.7: Sustainability

Sustainable HVAC design takes into account energy conservation in addition to environmental impact. In order to meet current sustainability criteria, team NUE Air's systems were designed in accordance with ASHRAE Standard 189.1 (see Section 2.3.4). Sustainability for each of the three system options was considered as part of the final decision matrix located in Section 10.0.

PV Louvers – Applicable to Options 1, 2, & 3
The decision to use a hybrid photovoltaic louver shading system allowed team NUE Air to take a novel approach in the design of the photovoltaic panels. With shading being one of the primary functions of the system, team NUE Air immediately turned its attention to the southern facade of the building, where sun exposure and high amounts of glazing combine to create a very problematic area from an HVAC design perspective. The initial panel sizing and placement was based upon the idea of providing shading to the large curtain walls on the East side of the Southern facade.

The Colt Shadovoltaic Louver system was selected to be paired with the Sunpower E19 320 Solar Panel. The energy analysis program RETScreen was used to model the system. The proposed system was modeled as a horizontally-mounted panel with single axis rotation to reflect the louver's ability to rotate its panels along a fixed horizontal axis. This rotation enables controllers to track the sun's altitude, thus minimizing the angle of incidence. The targeted load would require panels covering 126m^2 (1356ft^2) of the southern facade, accomplished with the distribution of panels shown in Figure 4.4. This area was shown by RETScreen to produce an annual energy output of 53,700 kWh, or 19.7% of the annual building energy load. This 19.7% energy generation translates to 7 points of LEED Energy and Atmosphere credit 2: On-Site Renewable Energy (see Section 4.10).

The subsidization of the initial cost of the photovoltaic array makes the proposed panel and mounting feasible. While sandstorms can wreak havoc on many building systems and they are a durability concern for any photovoltaic system in the region, the chosen panel is guaranteed to withstand the impact of a 1 inch hail stone traveling at 50 mph, which is a force that far exceeds what the panel will undergo in collision with sand particles.

See Section 9.1 for a complete life cycle cost analysis of this photovoltaic system.

4.8: Reliability

Reliability is evaluated based on the longevity, dependability, and overall quality of the building's equipment and operations. Accommodating component failure by the use of redundancy is important in the design of a reliable system. The inclusion of equipment redundancies is used to allow for the entire system to continue operating even when one system component is in need of replacement or repair. However, excessive redundancy leads to excessive initial costs. Reliability of all system options is evaluated in Section 10.0 as a consideration in the system selection matrix.

4.9: Noise Control

The acoustical qualities of the selected HVAC system are important to ensure limited distractions to students and faculty. Important noise control concerns were analyzed including self-generated noise created by turbulent duct airflow, airborne noise transmission from mechanical equipment through the ductwork, and noise transmission from the shop to the adjacent spaces. Noise control design strategies were implemented in order to comply with background noise criteria provided in the OPR (see Table 4.2).

Table 4.2: Noise Criteria

Area	Sound
Office & Administrative Support Spaces	NC 35
Classroom & Study Spaces	NC 30
Library	NC 30
Special Instruction Spaces	N/A

(source: Owner's Project Requirements)

Owner-required NC levels of 30 and 35 were achieved for each acoustically sensitive spaces given in Table 4.2 through use of duct lining, turning vanes, and low supply air velocity in the supply duct system. Trane Acoustics Program (TAP) analysis was used to select a duct assembly that both minimized duct borne noise carried from the heat pumps to the acoustically sensitive spaces as well as self-generated noise created by duct air turbulence. The shortest heat pump to diffuser path was analyzed in order to represent an acoustical worst case scenario. Rectangular lined duct was found to provide adequate attenuation, while the use of turning vanes and 2'' corner radiuses in the elbows and junctions was found to minimize regenerated noise (see Appendix E1.1 and E1.2). Using the chosen duct assembly a supply velocity of 350ft/min was chosen to meet the NC 35 acoustic criteria of the office and administrative spaces, while a supply velocity of 300ft/min was chosen to meet the NC 30 acoustic criteria of the classrooms, study spaces, and library (see Figures 4.7 and 4.8).

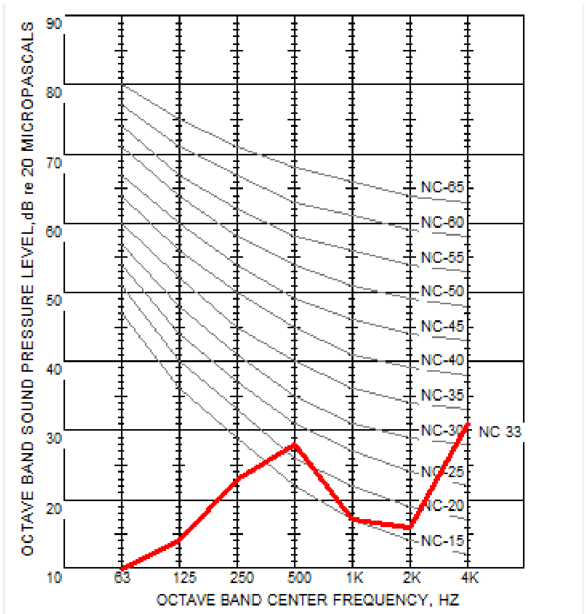


Figure 4.7: NC curve for space air supplied at 350ft/min (source: Trane Acoustics Program 2012)

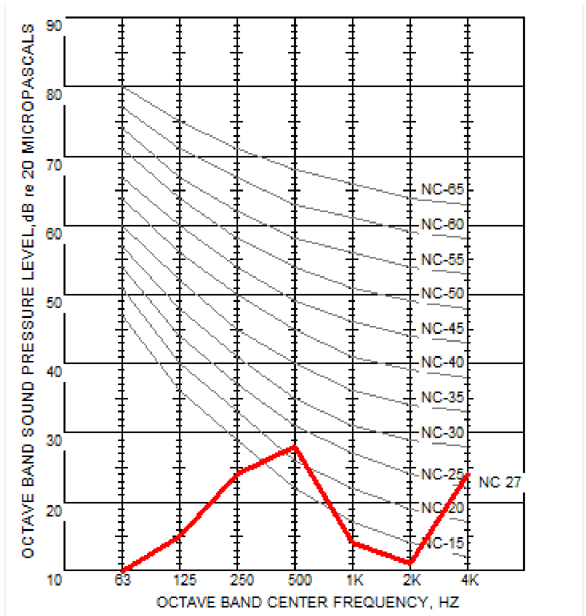


Figure 4.8: NC curve for space air supplied at 300ft/min (source: Trane Acoustics Program 2012)

Minimum sound transmission from the shop to the nearby acoustically sensitive offices on the first floor was achieved through selection of an appropriate STC shop wall assembly. Octave band sound levels within the shop were estimated based on measured decibel level noise data from woodworking equipment, where a total shop noise level was approximated from the logarithmic addition of 5 table saws and 3 large lathes. Representing this total dB level as a single sound source located in the center of the shop floor, as seen in Figure 4.9, allowed for TAP to be used to predict sound transmission between the shop and nearby offices. Typical wall constructions expected to be used in the pertinent spaces were chosen for analyses: a steel stud 5/8'' single gypsum board assembly for the office partition, and a 6"x8"x18" concrete block masonry wall assembly for the shop partition. Results showed that background noise levels on the order of NC 35 could be satisfied within the offices given the chosen wall assemblies, therefore no specialty wall assembly of high STC was needed (see Appendix E2.1). However, special gasketed sound proof doors are recommended for the shop wall partition in order to minimize sound leakage from the shop.

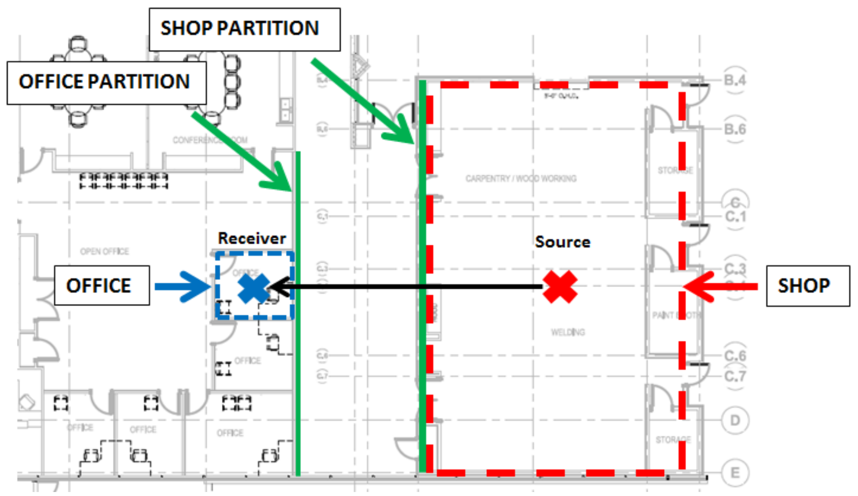


Figure 4.9: Sound Transmission source and receiver spaces defined

4.10: LEED

Though the owner did not specify that the building be evaluated with a building certification program, Team NUE Air investigated the potential Leadership in Energy and Environmental Design (LEED) points that each system option could achieve as an additional indicator of the environmental soundness of each option. This will also allow the owner to pursue LEED certification in the future if desired.

LEED is a green building certification program that recognizes buildings that have been designed, built, and operated in an environmentally-friendly manner. Building projects must meet prerequisites and earn credits toward LEED that are set in place by the United States Green Building Council (USGBC). Compliance with ASHRAE Standard 189.1 partially fulfilled most LEED requirements for this building. Table 4.3 summarizes the points earned by each system option that is outlined in Section 8.

Table 4.3: LEED Points

Applicable Credits		**Possible Points**	**VAV AHU**	**Fan coil units**	**Seawater Loop Field**
EAc1	Optimize Energy Performance	19	3	9	17
EAc2	On-Site Renewable Energy	7	7	7	7
EAc3	Enhanced Commissioning	2	2	2	2
EAc4	Enhanced Refrigerant Management	2	2	2	2
EAc5	Measurement and Verification	3	3	3	3
IEQc1	Outdoor Air Delivery Monitoring	1	1	1	1
IEQc3.1	Construction Indoor Air Quality Management Plan - During Construction	1	1	1	1
IEQc3.2	Construction Indoor Air Quality Management Plan - Before Occupancy	1	1	1	1
IEQc5	Indoor Chemical and Pollutant Source Control	1	1	1	1
IEQc6.2	Controllability of Systems - Thermal Comfort	1	1	1	1
IEQc7.1	Thermal Comfort - Design	1	1	1	1
IEQc7.2	Thermal Comfort - Verification	1	1	1	1
	Total	**40**	**24**	**30**	**38**

5.0: Building Design

5.1: Orientation

Building orientation was a key variable taken into consideration in order to minimize the solar heat gain through the glass panels on the structure's exterior. The most efficient way to reduce this heat gain was to minimize east and west direct exposure and maximize north and south exposures. It was determined that the side of the building with the entrance and the large amount of spandrel glass will face the south. This orientation will be ideal for the use of the photovoltaic panels requested in the OPR. Another LEED credit is possible if the east-west lengths of the building are equal to or greater than the north south lengths, and the east-west axis is within 15 degrees of geographic east-west. This is the planned orientation for the building so a LEED credit is achievable. Figure 5.1 shows the required LEED orientation.

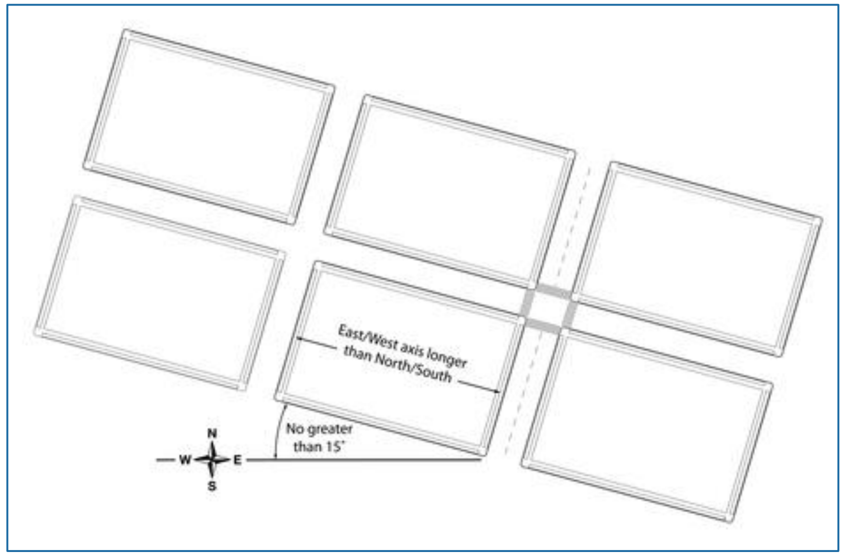


Figure 5.1: Required LEED orientation
(source: US Green Building Council)

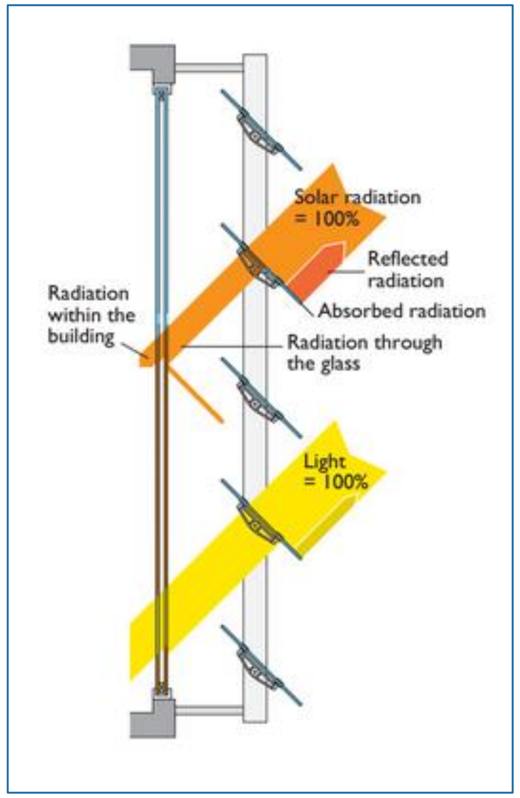


Figure 5.2: Louver shading system
(source: Colt Solar Shading Systems)

5.2 Shading

The louver system described in Section 4.4 provides solar shading to the southern façade of the building. Team NUE Air selected a louver system that meets the owner's requirements and provides an effective shading solution to help control solar load. The solar heat gain will be reduced which reduces a portion of the cooling load of the building. The louver system is controllable so the louvers track the position of the sun and automatically adjust to optimize the reduction in solar heat gain. Figure 5.2 shows a basic diagram of the louver system from the manufacturer that demonstrates how the louvers transmit daylight while reflecting and absorbing infrared solar radiation.

5.3 Envelope Assumptions

The ASHRAE standard wall construction and windows specified in the OPR were used in the design of the mechanical systems for this project. Team NUE Air understands that with the implementation of a more advanced wall construction and window design, the 15% energy saving could have been reached. However, Team NUE Air believes the spirit of this competition is to achieve the 15% savings through system selection, so effort was focused there.

6.0: Load Calculations

6.1: Modeling

The Trane TRACE 700 Load Design program was used to model the building. TRACE models energy and also the economic impact of design decisions. A total of four models were created: one for the Baseline and one for each system option. These models were utilized throughout the design process. During the preliminary design phase, a basic block load was calculated, treating the structure as a large space with simplistically modeled features (i.e. windows, doors, walls), which would give a general scale of the types of loading that would be placed on the school (see Figure 6.1). In addition, verification of the reasonableness of this preliminary model was made easy due to its simplicity, which allowed it to serve as a quick-check tool for the team throughout the rest of the design.

A load calculation model was created to find the peak load situation and therefore was used to size all system equipment. This model contained representations of each room in the building with all construction-related variables representative of those presented to team NUE Air in the OPR. Reasonable assumptions were made and verified with industry mentors where required information was not provided in the OPR. For example, the U-values of all curtain walls were found through hand calculations and entered into TRACE (see Appendix D.3). The TRACE load calculation showed a peak cooling load of 165 tons.

A second large contributor to building load is internal lighting load. This was explicitly outlined in the design criteria, which required the team NUE Air to research, assume, and justify the lighting load in the building. In a climate such as Doha, heat mitigation is as important an issue as low power usage. Thus, a 4000 lumen Lithonia 2VTL LED fixture was selected as a reference due to its highly efficacious design. Using luminaire placement at 10’ on center (O.C.) for general spaces and classrooms; and 6’ O.C. spacing in smaller offices, team NUE Air concluded that 0.7 W/ft^2 served as a conservative lighting heat gain load when averaged across the building.

TRACE checksums were created from the load calculation model and used to size the main system components.

Next, Team NUE Air created an energy model. Schedules that were supplied in the OPR were applied to the Load Calculation Model as to more realistically represent the loads experienced by the building in day-to-day function. The three system options were also fully modeled and, in combination with the utilities outlined in the design documents, the energy model was used to compare the energy and economic performance of the key systems.

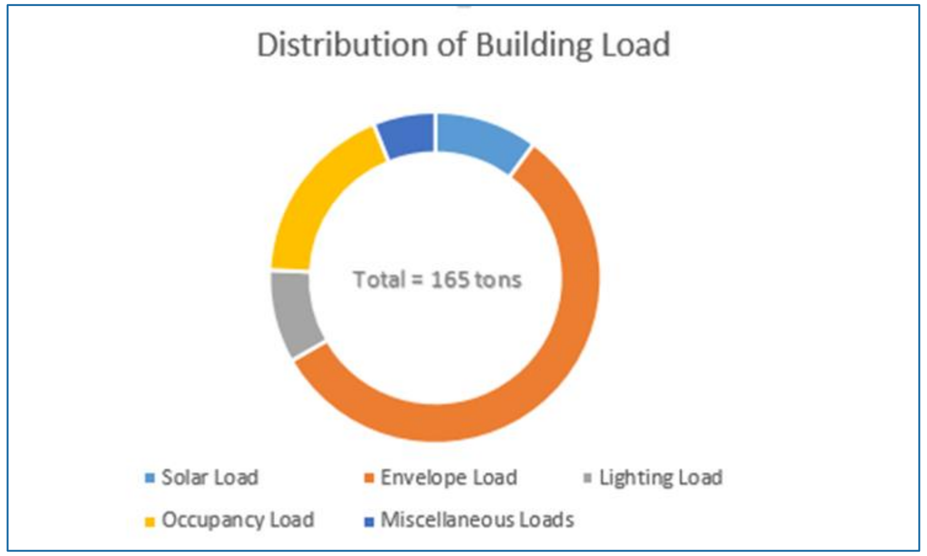


Figure 6.1: Distribution of Building Load
(source: authors)

6.2: Zoning

Occupant comfort has long been the key design factor in mechanical system design and selection, and the ability to precisely control the temperature of a conditioned space is one of the most effective methods in maintaining occupant satisfaction. By dividing each floor into multiple thermostat-specific zones, the temperature can be more accurately controlled on a per-room basis, allowing users to customize spaces to his or her liking.

Zoning criteria primarily includes space function, exterior exposure, and orientation; therefore, each classroom and large gathering space was placed in an individual zone to effectively cope with the load variations that can occur within a school throughout the day. Some exceptions to this are the stairs, where each of the two stair columns are zoned together. The stair columns and the elevator mechanical room are not zoned with larger adjacent rooms in order to avoid ventilation contamination during fire/smoke events (outlined in ASHRAE Standard 90.1 Section 6.4.3.4.1). Due to southern exposure and high amounts of glazing, the three exterior offices on the first floor (as seen in Figure A.1 of Appendix A.1) were placed in an individual zone separate from the other office spaces. To further increase control in these offices, special self-modulating diffusers were used on the supply side of the system. These diffusers each include their own motorized damper that allow the occupants to closely control the airflow in the office, which is key in a small room with high sensible load variation due to large southern-exposed windows. Despite being interior rooms, self-modulating diffusers were also added to the two additional offices to increase the level of control. See Appendix A.1 for the zoning diagrams of each floor.

6.3: Pressurization

As per standard design practice, all general spaces were designed to maintain positive pressure. Exfiltration is maintained with positive pressure, minimizing the amount of untreated outdoor air entering the space. This untreated infiltrating air can not only lead to space temperature variations, but more importantly it can affect the humidity of the space. This added humidity in the space occurs downstream of the air treatment systems, which means it cannot sufficiently treated, which can cause discomfort to the building occupants. Positive pressure is achieved throughout all general spaces in the building by providing more supply air to rooms than what is being removed from a room via exhaust and return systems. A 10% oversupply was chosen for this building based on industry standard.

Pressure differentials are used in the construction wing (special instruction area) of the building which includes the welding, woodworking, and painting spaces. These three spaces create contaminants in the air (e.g. wood particulates, welding fumes, paint overspray) that must be exhausted from the building without reaching other surrounding spaces. In order to create negative relative pressure, more air is removed from a specific room than what is supplied. To increase building efficiency, the overall supply air in the construction wing of the first floor is decreased, allowing intake of corridor air via motorized dampers located in the west wall of the construction area.

The total exhaust rate for the welding benches located in the construction room is calculated as:

$$R_{e,1} = (350\frac{cfm}{ft.})(length)$$
$$R_{e,1} = (350\frac{cfm}{ft.})(4ft.)$$
$$R_{e,1} = 1{,}400\,cfm$$

Figure 6.2: Welding hood
(source: Industry Documentation)


Figure 6.3: Cone hood flexible connection
(source: Industry Documentation)


Figure 6.4: Donaldson Torit filter unit
(source: Industry Documentation)

The total exhaust rate for the cone hood flexible exhaust connections located in the construction room is calculated as:

$$R_{e,2} = (600\frac{cfm}{connection})(n_{connection})$$
$$R_{e,2} = (600\frac{cfm}{connection})(6)$$
$$R_{e,2} = 3{,}600\,cfm$$

The total exhaust rate for the construction room is the summation of the above rates:

$$R_{tot} = R_{e,1} + R_{e,2}$$
$$R_{tot} = 1{,}400cfm + 3{,}600cfm$$
$$R_{tot} = 5{,}000cfm$$

(source: ACGIH)

Air-to-cloth ratio is a typical form of rating for dust collectors. This ratio represents the volumetric flow rate of air flowing through a collector over the total area of cloth filter.

A baghouse dust collector with an air-to-cloth ratio between 6.0-8.0 will be used in the woodworking area. The selected Donaldson Torit unit shown in Figure 6.4 has a filter area of 450 square feet, and with a required airflow of approximately 3,100 CFM it is apparent that unit falls within the acceptable ratio and is properly sized:

$$AtC\ Ratio = \frac{3{,}100CFM}{450ft^2}$$
$$AtC\ Ratio = 6.9\frac{ft}{min}$$

6.4: Baseline

Team NUE Air developed a Baseline system within TRACE to create a model to which each system option could be compared. A Baseline system consisting of a packaged variable air volume DX cooling rooftop unit, with variable air volume hot water reheat terminal units was used. All modeling complies with the Performance Rating Method described in ASHRAE Standard 90.1, Appendix G. The Baseline has a total building energy consumption of 2,558,000 kBtu/year.

7.0: Major System Components

7.1: VAV Air Handling System

Variable air volume systems (VAV) provide varying rates of conditioned room air to each zone controlled via a motorized damper. All air handling unit fans are also equipped with variable frequency drives that modulate according to the needs of the zones. These two areas of controllability provide a substantial increase in efficiency over a constant volume system. Each VAV box includes electric reheat which is used when the damper in the VAV box reaches its minimum airflow position in cooling, but the zone still requires heating. Each air handling unit is coupled with an energy recovery ventilator (ERV) in the form of an total enthalpy wheel. The enthalpy wheel was chosen because it is the least expensive yet most efficient energy recovery system due to its simple design. Each air handling will also be equipped with UV lights for germicidal radiation located 12 inches off of the cooling coils to keep the pressure drop stable and maintain a healthy and clean environment. The air handling units are equipped with special sand louvers which are commonly used in desert areas to prevent sand entrainment.

7.2: Fan Coil Units with Dedicated Outside Air System

Fan coil units (FCU) use a heating and/or cooling coil coupled with a fan to deliver conditioned air to a space. When space temperature increases, the thermostat signals the unit to begin circulating chilled water through its cooling coil. Air is blown over the coil, thus cooling the air. Fan coil units are located externally to each space, utilizing ducted runs to mitigate noise issues.

Since fan coil units are stand alone in nature, they are unable to bring outdoor air into the building under their own power, requiring the implementation of a dedicated outdoor air system (DOAS). The DOAS contains its own heating and cooling coils in order to pretreat the ventilation air, which reduces the load on the fan coils within the building. In climates such as Doha, a cooling coil arrangement can be utilized within the DOAS in order to adequately dehumidify the outdoor air. This DOAS provides an outdoor air amount that is compliant with ASHRAE Standard 62.1. Additionally, the DOAS utilizes an ERV enthalpy wheel to precondition the outside air using energy from the relief air to decrease wasted energy.

7.3: Water Source Heat Pump System with Dedicated Outside Air System

Water source heat pumps utilize a refrigeration cycle to heat or cool the supply air. Water source heat pumps use a common building-wide water loop for heat transfer through a water-to-refrigerant heat exchanging process. By being able to simultaneously reject and take on heat from this water loop, the heat pump system can be self-regulating, meaning that an equal amount of heat is being added (usually from interior spaces) and removed (usually from exterior spaces) from the system, making it extremely efficient. When outdoor conditions require perimeter cooling as will usually be the case in Doha, additional heat can be rejected through a geothermal or seawater loop, where conditions are constant and predictable on a day-to-day basis.

A reversing valve dictates the direction of refrigerant flow based on the conditioning mode. In cooling mode, the heat pumps reject heat to the water loop. In heating mode, the water loop rejects heat to the heat pump, where it is then supplied to the space. Next, the refrigerant flows into an air-to-refrigerant coil. This provides the required heating or cooling for the supply air.

Similar to fan coil units, heat pumps do not bring outdoor air into the building. Thus, ventilation air is delivered to each space through the use of a DOAS in the same manner outlined in Section 7.2.

7.4: Thermal Ice Storage

The thermal ice storage system uses the chiller to form ice during off-peak utility rate hours that is then used as a source for cooling during on-peak utility rate hours. Per the OPR, on-peak utility rate hours occur from 9:00 a.m. to 7:00 p.m. Monday through Saturday. During these on-peak hours, the chilled water passes through the ice created in the thermal storage tanks during off-peak hours to lower its temperature before it enters the air handling unit. This reduces the amount of electricity required to condition the air. Thus, the cost of cooling for the building is reduced.

Thermal ice storage allows for humidity control due to the cold temperature of chilled water. Additionally, pumping is saved by lower temperature water with larger ΔT's.

Based on the TRACE calculations, a 100 ton ice-making chiller is required for the thermal storage system. Each thermal ice storage unit has a capacity of 9150 gallons and provides 761 ton-hours
of cooling.

7.5: Seawater Loop Field

The closed seawater loop system utilizes the Persian Gulf, located adjacent to Doha, as a heat sink or heat source. High-density polyethylene (HDPE) piping was chosen to complete the loop system due to its corrosion resistant properties. A slinky configuration of overlapping HDPE coils running along the floor of the gulf is distributed to the condensers in each water source heat pump by means of a plate and frame heat exchanger to reject heat from the condensers. The warmed water is then discharged back to the slinky coil submerged in the Persian Gulf. An environmentally safe corrosion inhibitor will be added to the loop water to ensure no harmful chemicals are introduced into the seawater.

8.0: Systems Considered

Option 1

8.1: VAV air handling system with electric reheat, a chiller with a cooling tower assisted by water fountain spray cooling tower, utilizing thermal ice storage

The primary system of this option employs a supplemental water fountain spray cooling tower, a primary cooling tower, and a water cooled chiller. As a part of cost-saving strategies, the thermal ice storage system outlined in Section 7.4 is utilized in this option as well, reducing peak loads and hence first cost for the primary chiller. The secondary side of this option employs an air handling unit with zone-level VAV boxes including electric reheat (see Section 7.1). A schematic of this system can be found in Appendix B.1. Utilizing TRACE, this system was found to have a total building energy consumption of 2,138,000 kBtu/year, making it **17%** more efficient than the Baseline model.

Advantages:
- VAV has low initial cost
- VAV is easily maintained
- VAV has individual zone temperature control, thus increased thermal comfort
- VAV allows for greater design flexibility
- Thermal storage generates chilled water less expensively during peak loads

Disadvantages:
- VAV may not always meet required ventilation air
- The noise associated with the airflow changes as a VAV box modulates could be distracting for students
- Thermal storage requires a significant amount of space and may not be visually pleasing

Option 2

8.2: Fan coil units with dedicated outside air system, a chiller with a cooling tower assisted by water fountain spray cooling tower, utilizing thermal ice storage

Similar to Option 1, the primary system of this option employs a supplemental water fountain spray cooling tower, a primary cooling tower, and a water cooled chiller. This option also utilizes the same thermal ice storage as Option 1, as outlined in Section 7.4. The secondary side of this option employs fan coil units with a DOAS system for the treatment of ventilation air, outlined in Section 7.2. The DOAS is a variable volume system that motorized dampers at the connection to each fan coil. This allows for energy reduction by decreasing ventilation air rates whenever the fan coil zones are unoccupied. This allows for the implementation of CO_2 sensors that control DOAS flow rate at the zone level. A schematic of this system can be found in Appendix B.2. Utilizing TRACE, this system was found to have a total building energy consumption of 1,831,000 kBtu/year, making it **28%** more efficient than the Baseline model.

Advantages:
- FCU has lower initial cost due to lack of ductwork
- Pipes to the fan coil units are smaller than ducts, thus reducing plenum size requirements
- DOAS is versatile, and easily paired with any system
- DOAS has low energy requirements
- DOAS controls humidity
- Thermal storage generates chilled water less expensively during peak loads

Disadvantages:
- Fan coil noise may be more apparent because they are located directly in the space
- Thermal storage requires a significant amount of space and may not be visually pleasing

Option 3

8.3: Water source heat pump system with dedicated outside air system utilizing a closed seawater loop field

The primary side of this option employs a closed seawater loop system (see Section 7.5). The secondary side of this option employs water source heat pumps (see Section 7.3). Water source heat pumps transfer the thermal energy from the condenser coils in the heat pumps to the seawater loop. Both return and supply ductwork connect directly to each heat pump in order to supply conditioned air to and return air from the space. The ventilation system for this option employs a dedicated outside air system. The outdoor air duct from the DOAS connects on the return side of the heat pump, allowing for mixing to occur before the air enters the unit. Similar to Option 2, the DOAS is a variable volume system that utilizes motorized dampers at the connection to each water source heat pump. Again, energy reduction is realized by decreasing ventilation air during unoccupied mode and CO_2 sensors and controls are easily applied at the zone level. A schematic of this system can be found in Appendix B.3. Utilizing TRACE, this system was found to have a total building energy consumption of 1,445,000 kBtu/year, making it **44%** more efficient than the Baseline model.

Advantages:
- Seawater loop field through the Persian Gulf is a sustainable energy source
- Seawater loop field is energy efficient
- Seawater loop field has low lifecycle cost
- DOAS is versatile, and easily paired with any system
- DOAS has low energy requirements
- DOAS controls humidity

Disadvantages:
- High initial cost
- Burying of pipes in the Persian Gulf
- Heat pumps can create noise issues due to internal compressors

9.0: Cost

9.1: Life Cycle Cost Analysis for each system

The life cycle cost of each system was calculated using a spreadsheet and then compared to values calculated in TRACE. The initial cost of the WSHP with seawater loop is higher than the other two systems, however the operation and maintenance costs are substantially less giving it a lower life cycle cost. Using present worth analysis and comparing the total cost per year, the WSHP with seawater loop was found to have a payback of 4 years compared to Option 1, and a payback of 13 years compared to Option 2. Savings from choosing WSHP with seawater loop are $695,000 (18.3%) over VAV with Ice Storage and $220,000 (5.8%) over FCU with Thermal Ice Storage over a 50 year analysis. Figures 9.1 and 9.2 show the life cycle cost of each system over 50 years.

All costs are calculated using present worth analysis and include the rate of inflation, return on investment, and utility rate increases as specified by the OPR.

9.2: Initial Cost

The initial cost for every system (besides the equipment specified within each option) includes HVAC ductwork, hydronic piping, and temperature controls for the entirety of the building (46,420 ft^2). A design contingency of 10%, a construction contingency of 5%, sales tax at 7%, and all labor costs were included as well.

9.3: Operating Cost

The operating cost for every system includes utilities for on-peak and off-peak electricity consumption ($/kWh), on-peak electricity demand ($/kW), city water ($/ft^3), and city sewer ($/gal).

9.4: Maintenance and Repair Cost

Two case studies in Chapter 37 of the 2011 ASHRAE Handbook HVAC Applications covering over 500 buildings outline the mean and median maintenance cost per square foot of buildings. From these studies, the maintenance costs were estimated. On average, heat pumps are more expensive to maintain than their FCU and VAV counterparts, which is accounted for in the annual maintenance cost per system. Chapter 37 also identifies the median service life of components, including heat pumps, VAV units, FCU's, chillers, and cooling towers. A survival curve for centrifugal chillers over a 50 year span gives an estimate for the survival of other components. This curve was used along with the median service life to identify how many units are likely to fail over time.

The annual maintenance cost for every system includes the cost to clean and maintain the photovoltaic array at a rate of $2,000 per year. The replacement cost for every system includes the present cost of $21,150 to replace the photovoltaic array at year 25. All other maintenance and replacement costs for each system vary depending on components used.

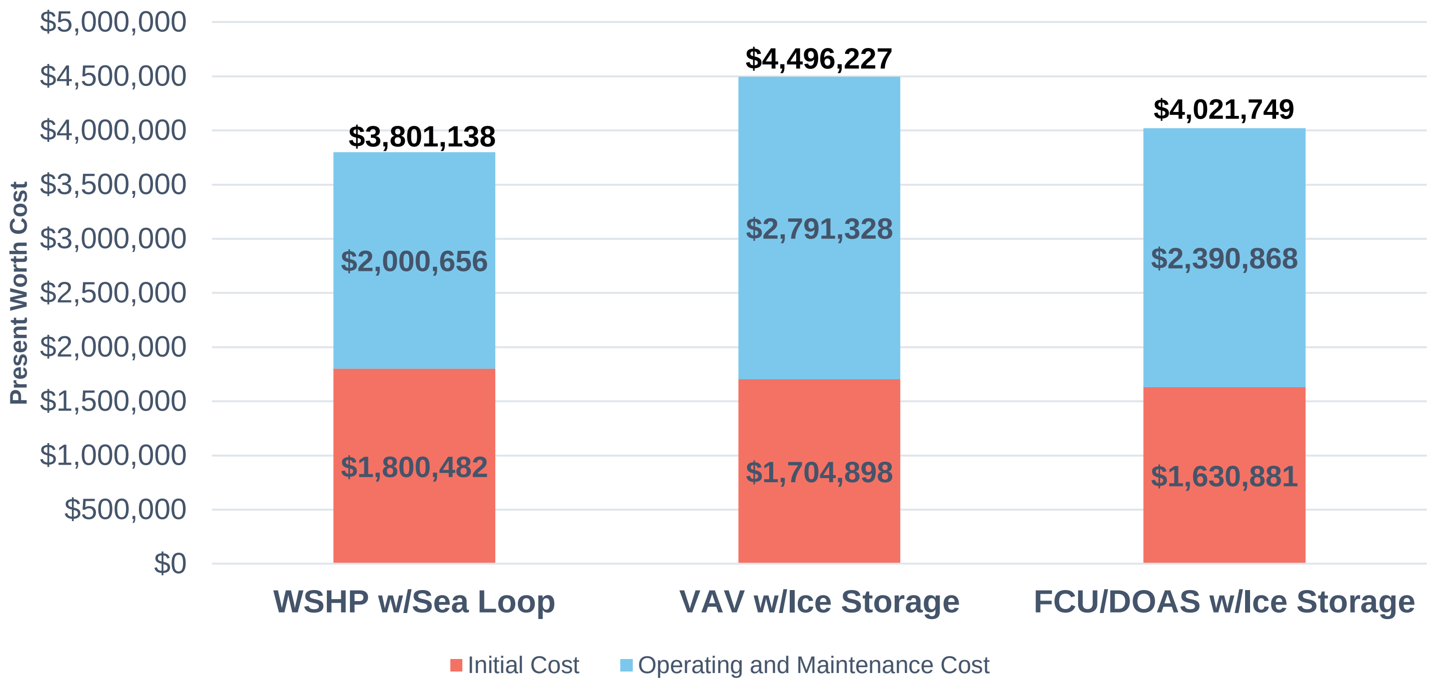


Figure 9.1: Initial, Operating, and Maintenance Cost

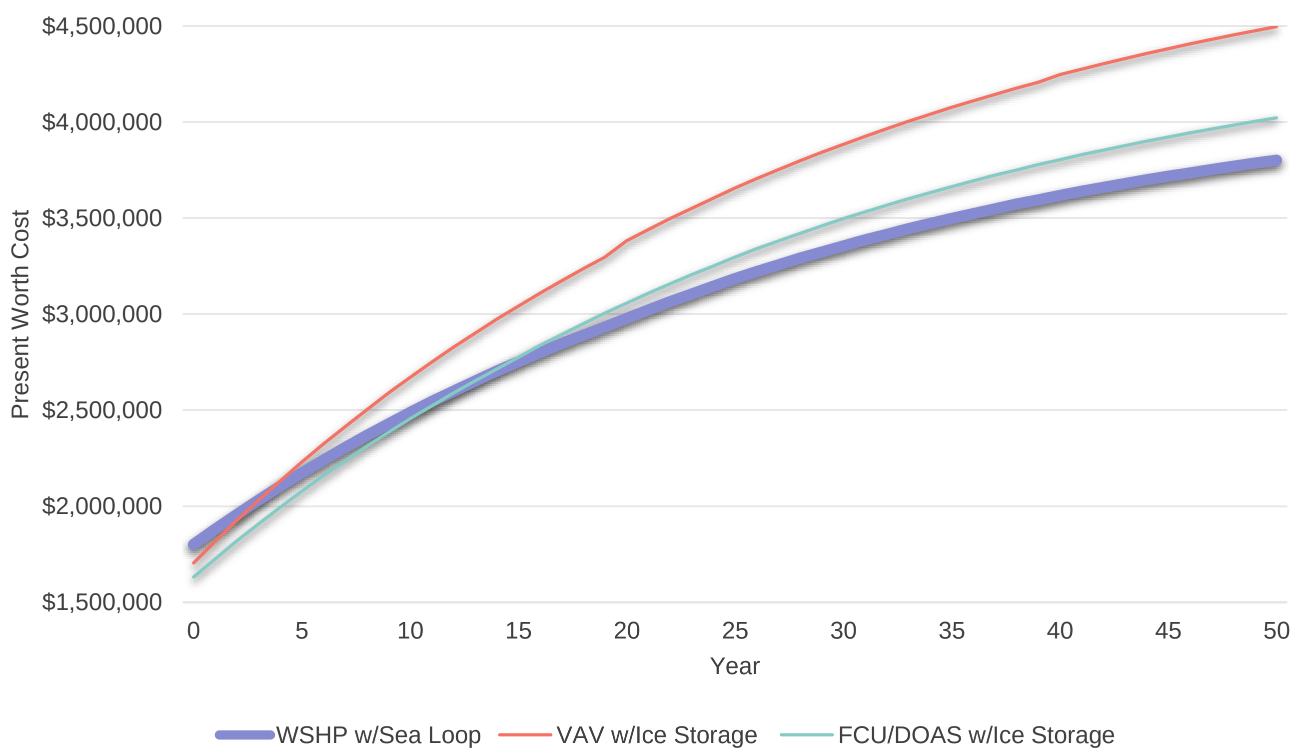


Figure 9.2: Life Cycle Cost

9.0: Cost

VAV with Thermal Ice Storage

Initial Cost + Annual Maintenance Cost + Replacement Cost + Utility Cost = Life Cycle Cost

$1.7M | $14,000 | $4,000-60,000 | $105,600 | $4.5M

FCU/DOAS with Thermal Ice Storage

Initial Cost + Annual Maintenance Cost + Replacement Cost + Utility Cost = Life Cycle Cost

$1.63M | $14,000 | $6,000-27,150 | $87,900 | $4.0M

WSHP with Seawater Loop

Initial Cost + Annual Maintenance Cost + Replacement Cost + Utility Cost = Life Cycle Cost

$1.8M | $17,000 | $9,000-30,150 | $67,900 | $3.8M

Methods of calculation:

AHU per CFM; VAV boxes, chillers, and thermal storage per each; Temperature controls, HVAC ductwork, and hydronic piping per square foot of the building. Other factors include sales tax at 7%, design contingency at 10%, construction contingency at 5%, and overhead and profit at 5%.

Annual utility cost including electricity, water, and sewer.

VAV with Thermal Ice Storage

The initial cost of the VAV with Ice Storage system is $1.7 million including 3 AHUs, 50 VAV units, a 100 ton water cooled chiller, and a 790 ton-hour thermal storage system. The annual maintenance cost was estimated to be $14,000 starting after the first year. In addition, the replacement cost of the chiller was estimated to be $60,000 at year 20 and 40 along with $4,000 replacement costs for VAV units starting in year 20.

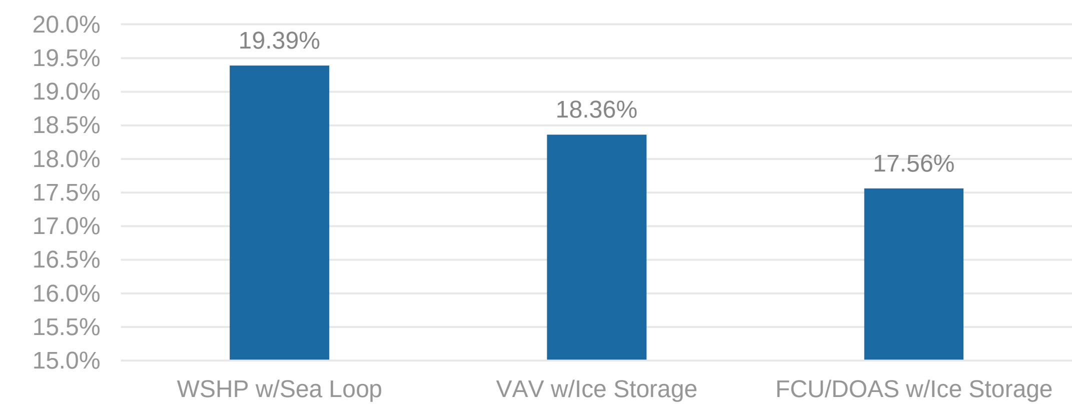
FCU/DOAS with Thermal Ice Storage

The initial cost of the FCU/DOAS with Ice Storage system is $1.63 million including 50 FCUs, an energy recovery ventilator supplying 12,000 CFM, a 100 ton water cooled chiller, and a 790 ton-hour thermal storage system. The annual maintenance cost was estimated to be $14,000 starting after the first year. In addition, the replacement cost was estimated to be $6,000/year starting in year 15.

WSHP with Seawater Loop

The initial cost of the WSHP with seawater loop system is $1.8 million including 50 WSHP units (starting at half ton), an energy recovery ventilator supplying 12,000 CFM, and 77,400 feet of geothermal sea loop. The annual maintenance cost was estimated to be $17,000 starting after the first year. In addition, the replacement cost was estimated to be $9,000/year starting in year 20.

Initial Cost of System as a Percent of Total Building Budget



Cost Per Square Foot

	WSHP w/Sea Loop	VAV w/Ice Storage	FCU/DOAS w/Ice Storage
HVAC $/ft²	$38.78	$36.72	$35.13
%HVAC of Total Building Budget	19.39%	18.36%	17.56%

10.0: Decision Matrix

10.1: Decision Matrix Description

Team NUE Air created a decision matrix in order to select an HVAC system for the building that fulfills all design goals (see Figure 10.1). This matrix includes values on a scale from 0 to 5 (0 as worst, 5 as best) for each system type in a variety of categories and each category is weighted overall. The matrix is divided into plant systems and distribution systems to allow a more detailed analysis of individual components for each system option.

		Initial Cost	Operating Cost	Temp Control	Humidity Control	Controll-ability	Indoor Air Quality	System Reliability	Design Flexibility	Spatial Requirement	Noise	Sustain-ability	High Performance Building Compliance	Maintenace Requirement	Synergy with Architecture	Total
Plant Systems	**Category Weighting**	20%	20%	n/a	n/a	6%	n/a	12%	9%	3%	3%	12%	6%	6%	3%	
	Air Cooled Condenser	5	1	n/a	n/a	3	n/a	2	3	4	1	1	3	2	2	50%
	Water Cooled Condenser	3	3	n/a	n/a	3	n/a	3	3	3	2	4	4	2	2	61%
	Ground Source Condenser	1	5	n/a	n/a	3	n/a	4	3	3	5	4	4	3	3	67%
	Non-condensing Boiler	4	2	n/a	n/a	2	n/a	4	3	3	3	2	2	3	3	58%
	Condensing Boiler	2	4	n/a	n/a	4	n/a	3	3	3	3	4	3	3	3	64%
	District Energy	5	1	n/a	n/a	2	n/a	5	4	5	5	3	2	5	4	70%
Distributed Systems	**Category Weighting**	20%	20%	3%	3%	3%	6%	9%	9%	3%	3%	12%	3%	3%	3%	
	Constant Volume	3	3	2	3	2	3	3	3	4	3	3	1	3	2	58%
	Variable Volume	2	3	5	3	5	4	3	3	3	2	3	3	3	3	59%
	Many Air Source Heat Pumps	2	4	4	3	4	2	4	4	2	3	3	4	4	3	64%
	Many Water Source Heat Pumps	3	4	4	4	4	3	4	4	3	4	4	4	4	4	74%
	Fan Coil Unit	4	3	4	1	3	1	3	3	3	1	2	3	4	3	58%
	Displacement Ventilation	2	4	3	1	3	4	3	2	2	4	2	3	3	3	56%
	Energy Recovery Ventilator	2	4	4	3	3	4	3	4	3	4	3	5	4	4	67%

Figure 10.1: Decision Matrix

11.0: Selected System

11.1: System Description

After comparing all HVAC system options against the Baseline and each other, team NUE Air chose the water source heat pump system utilizing a closed seawater loop field as the optimal system for the building. This system fulfills all design goals established by team NUE Air. The system has the lowest life cycle cost of the three systems considered. The seawater loop field provides a sustainable energy source with a low environmental impact. Comfort and indoor environmental quality are easily maintained due to the adjustability of the system within the building. The system is a high-performing, sustainable, low-carbon design that is easily integrated into the building, making it architecturally-pleasing. As evidenced by the analysis conducted by team NUE Air, the system provides a highly-efficient, sustainable, and reliable option for the Doha Junior College.

12.0: Conclusion

12.1: Recommendation

The water source heat pump system with dedicated outside air system utilizing a closed seawater loop field is the optimal system selected. Team NUE Air recommends this system for implementation to the new junior college in Doha, Qatar. While all three system options are viable for the building, this water source heat pump with seawater loop field system scored the highest on the decision matrix and provides the most suitable design for both the building's and the Owner's requirements.

This system showed a 44% energy improvement over the Baseline. This system is projected to cost \$3.8 million over a 50 year period, which is \$695,000 less than the VAV with thermal ice storage option and \$220,600 less than the FCU/DOAS with thermal ice storage option.

Team NUE Air chose the water source heat pump system with dedicated outside air system utilizing a closed seawater loop field as the optimal system because it encompasses elements of efficiency; occupant health, safety, and comfort; and sustainability in a single, maintainable design that has a low life cycle cost.

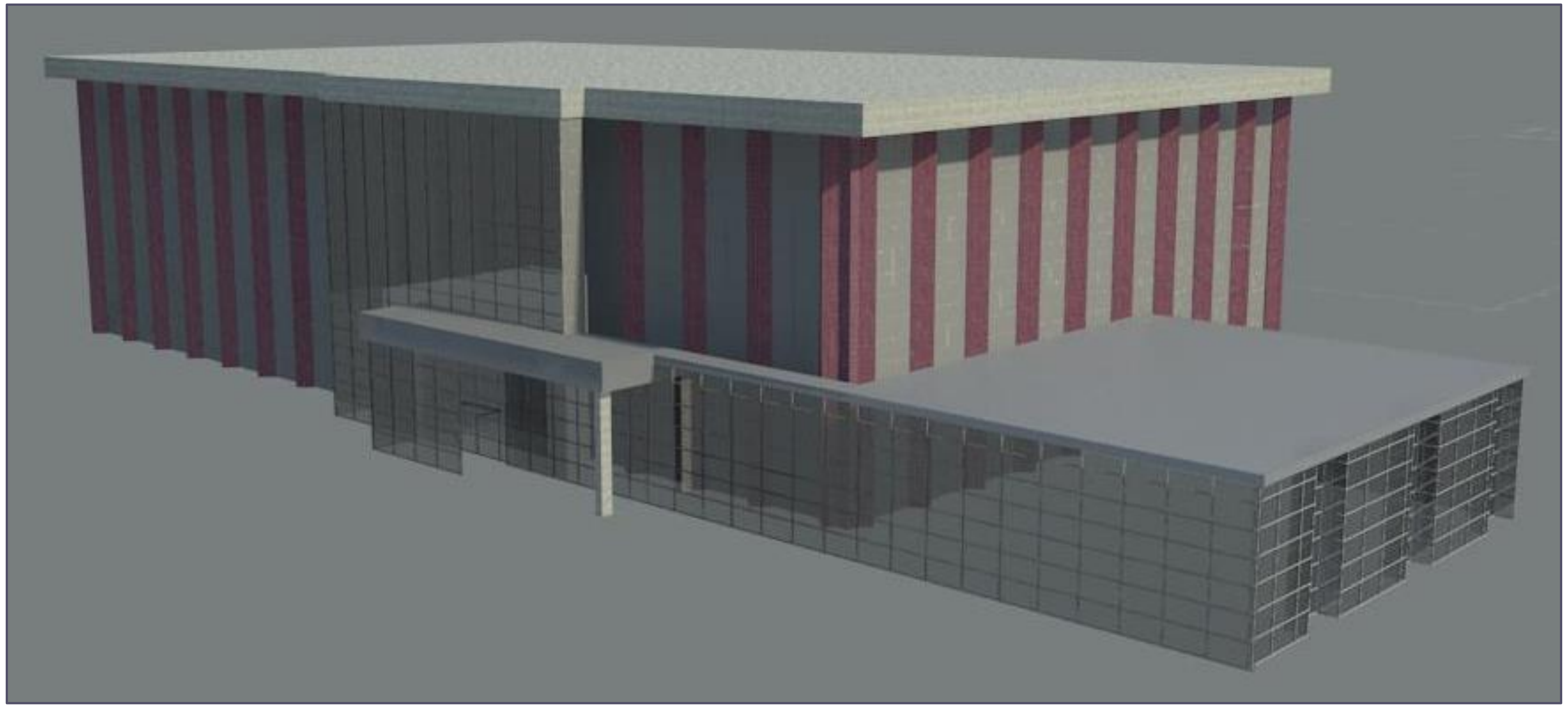


Figure 11.1: Junior College in Doha, Qatar
(source: authors)

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Appendix A: Zoning Diagrams

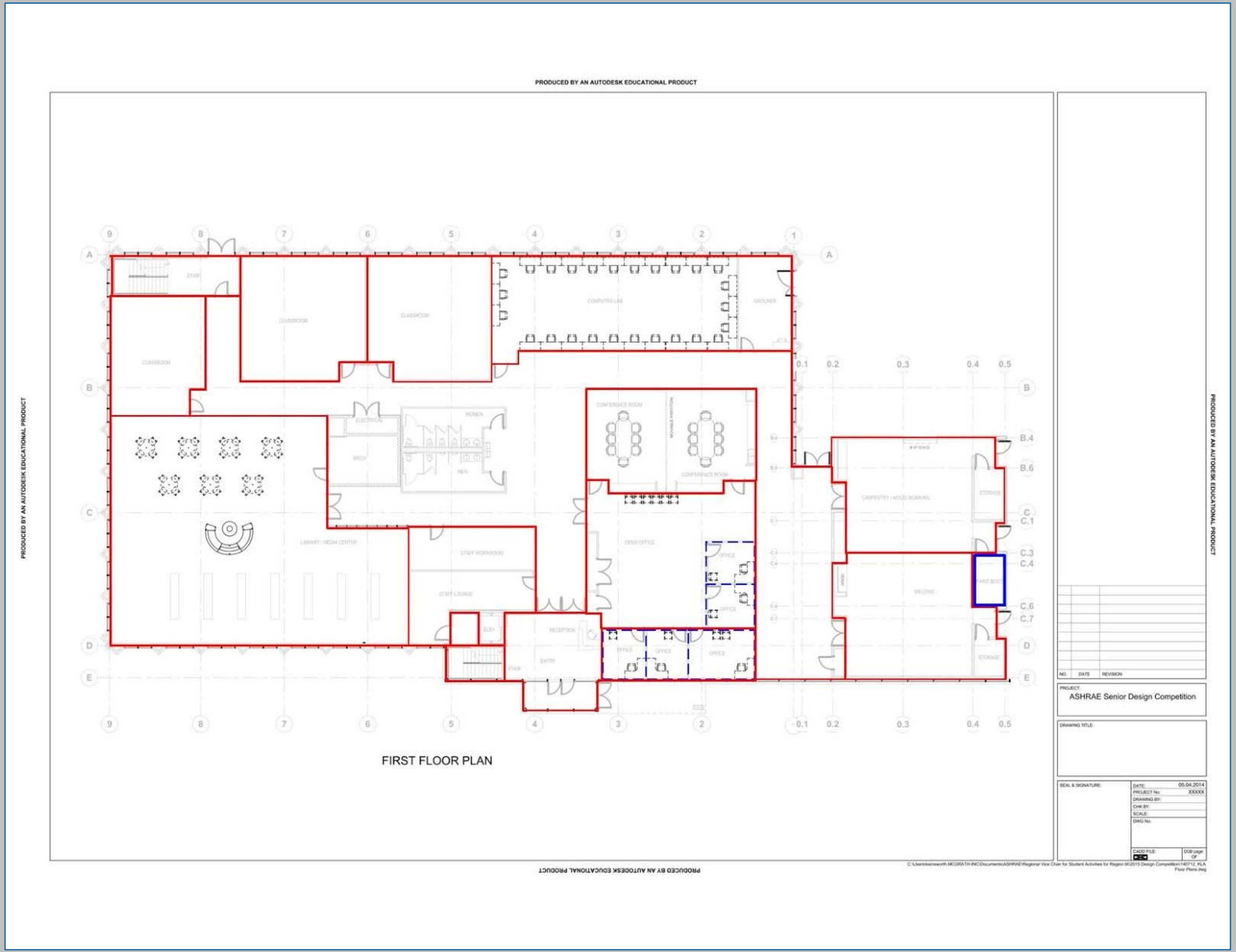


Figure A1.1: First Floor Zoning Diagram

Figure A1.2: Second Floor Zoning Diagram

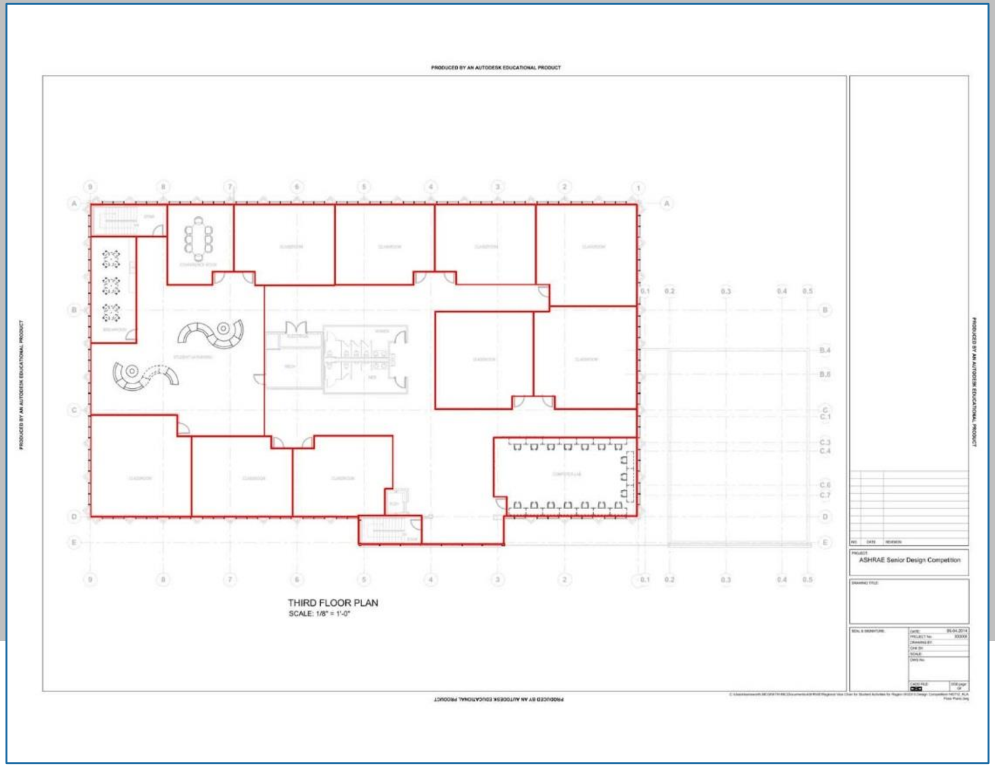
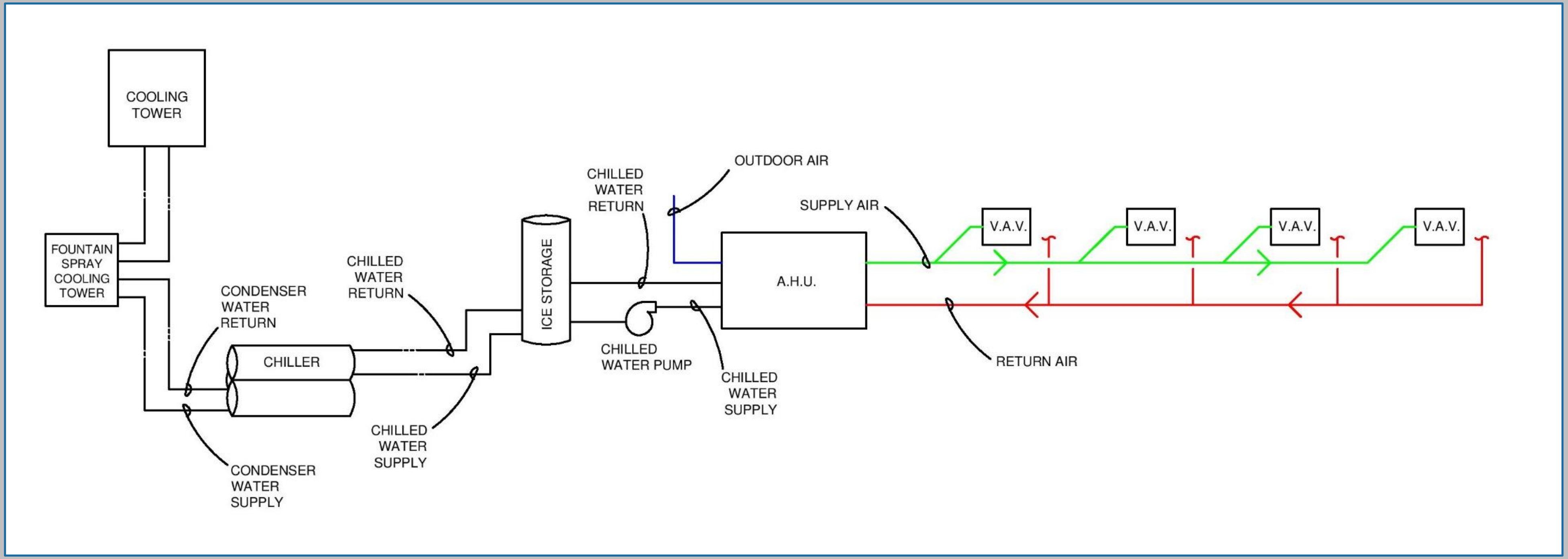


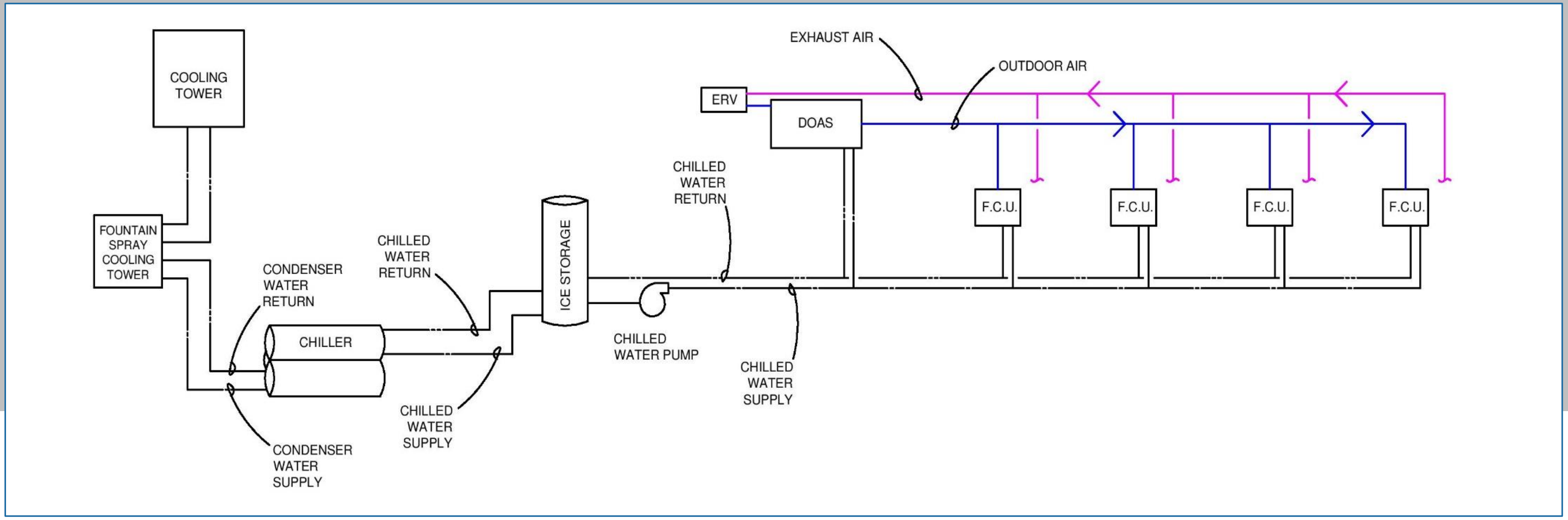
Figure A1.3: Third Floor Zoning Diagram

Appendix B: System Schematics

B.1: Option 1 – VAV Air Handling System Utilizing Thermal Storage

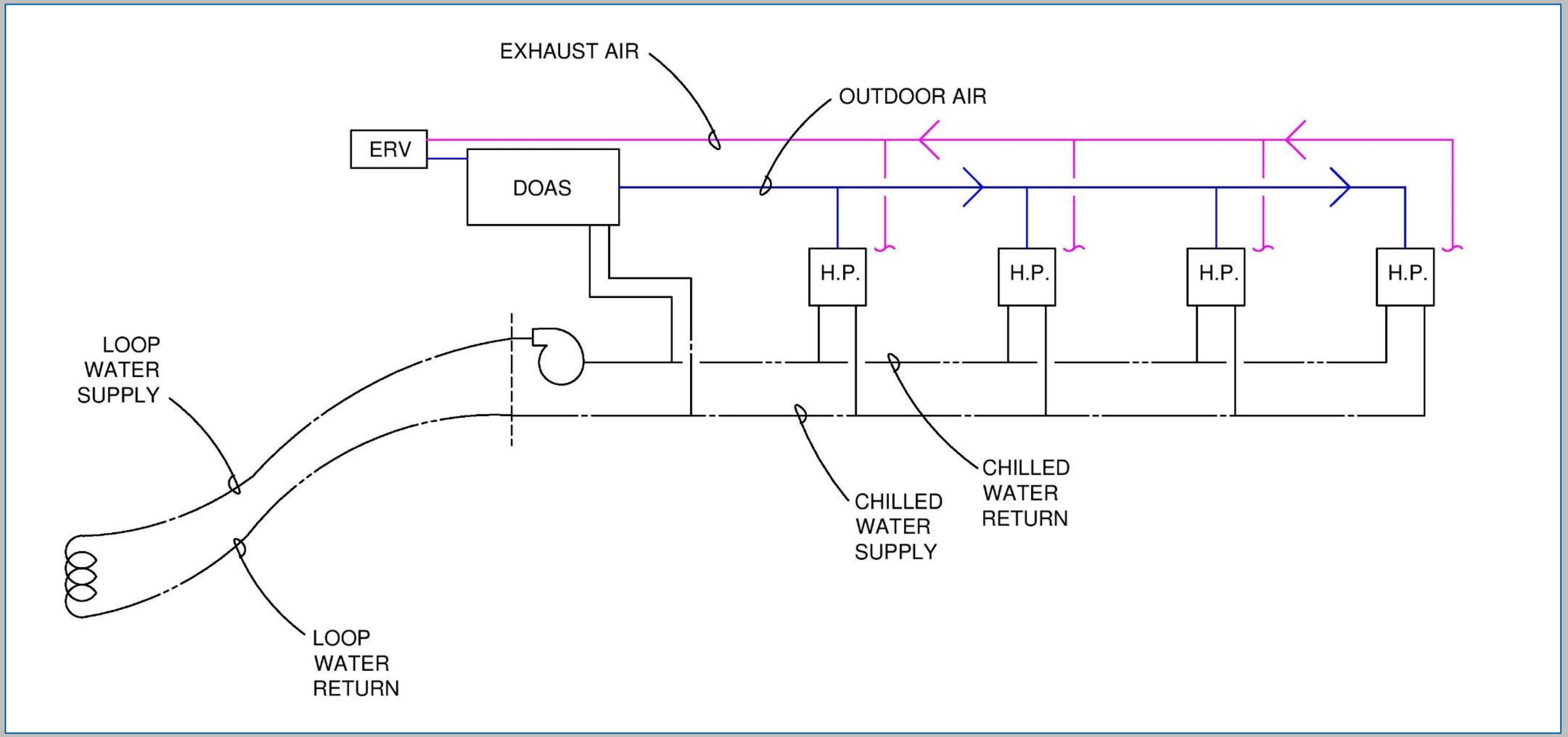


B.2: Option 2 – Fan Coil Units With Dedicated Outside Air System Utilizing Thermal Ice Storage



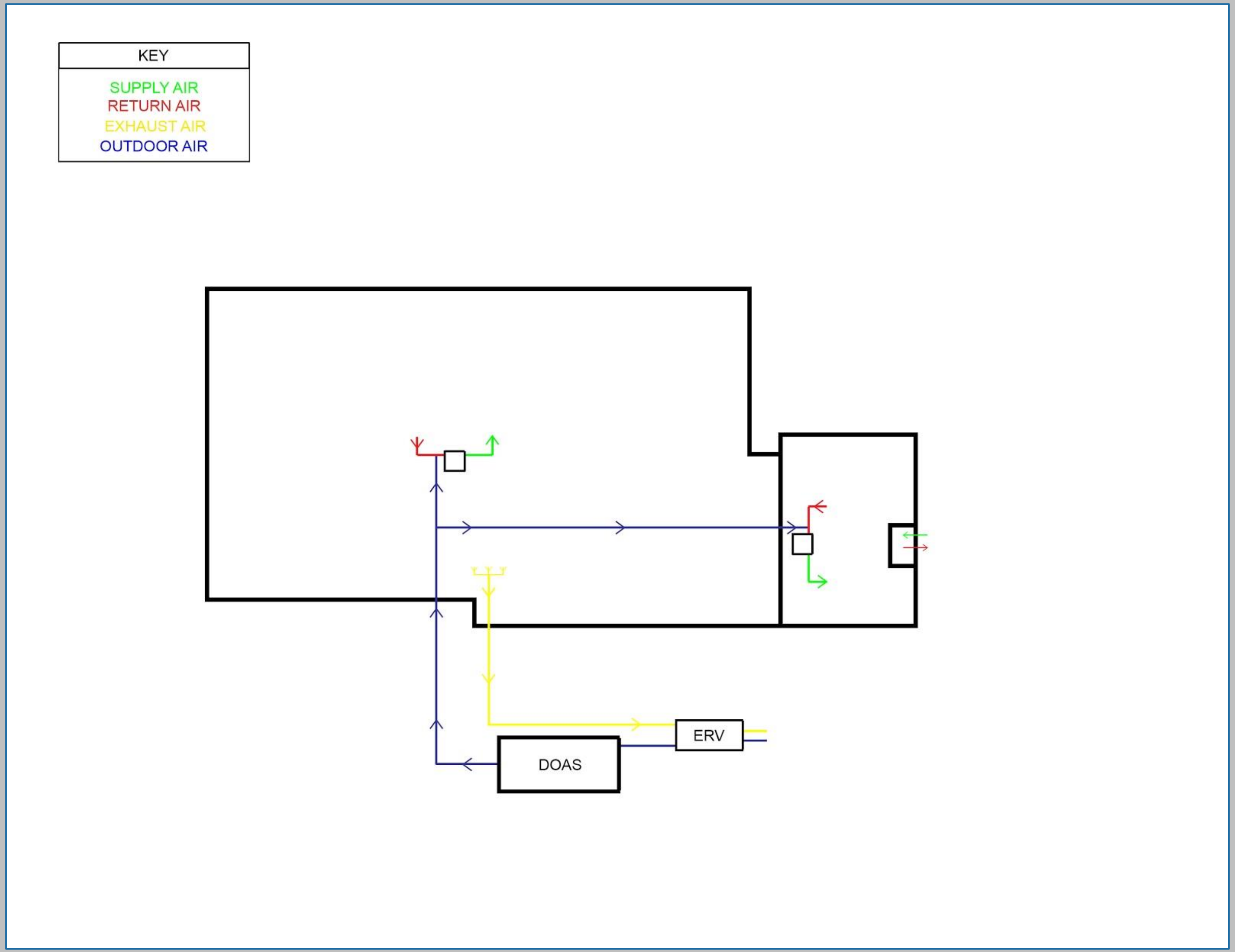
Appendix B: System Schematics

B.3: Option 3 – Water Source Heat Pump System Utilizing a Closed Seawater Loop Field

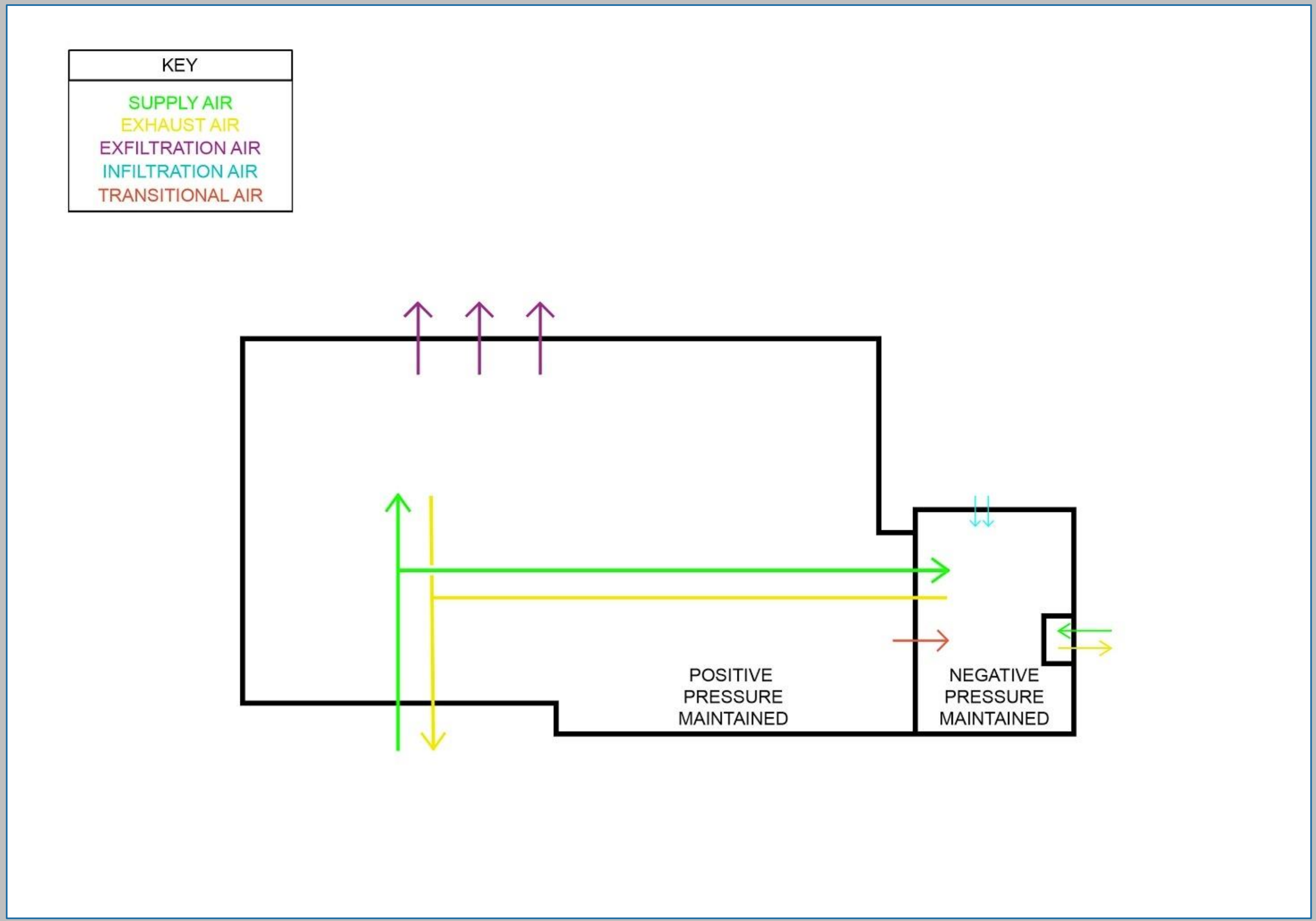


Appendix B: System Schematics

B.4: One-Line Diagram



B.5: Pressurization Diagram



Appendix C: Life Cycle Cost Analysis

Life Cycle Cost for VAV w/Ice Storage

Year	Utility Cost	Annual Maintenance Cost	Replacement Cost	Initial Cost	Present Worth
0				$1,704,898	**$1,704,898**
1	$105,609	$14,420			**$112,177**
2	$109,238	$14,853			**$108,386**
3	$112,993	$15,298			**$104,724**
4	$116,877	$15,757			**$101,186**
5	$120,895	$16,230			**$97,768**
6	$125,052	$16,717			**$94,467**
7	$129,353	$17,218			**$91,277**
8	$133,803	$17,735			**$88,196**
9	$138,406	$18,267			**$85,219**
10	$143,168	$18,815			**$82,344**
11	$148,095	$19,379			**$79,566**
12	$153,192	$19,961			**$76,882**
13	$158,466	$20,559			**$74,289**
14	$163,922	$21,176			**$71,784**
15	$169,566	$21,812			**$69,364**
16	$175,406	$22,466			**$67,026**
17	$181,448	$23,140			**$64,767**
18	$187,699	$23,834			**$62,585**
19	$194,166	$24,549			**$60,476**
20	$200,857	$25,286	$102,400		**$84,902**
21	$207,779	$26,044	$6,520		**$58,046**
22	$214,941	$26,825	$6,640		**$56,069**
23	$222,351	$27,630	$6,760		**$54,159**
24	$230,018	$28,459	$6,880		**$52,314**
25	$237,950	$29,313	$22,863		**$53,455**
26	$246,156	$30,192	$7,120		**$48,812**
27	$254,647	$31,098	$7,240		**$47,150**
28	$263,432	$32,031	$7,360		**$45,545**
29	$272,521	$32,992	$7,480		**$43,995**
30	$281,925	$33,982	$7,600		**$42,498**
31	$291,655	$35,001	$7,720		**$41,052**
32	$301,722	$36,051	$7,840		**$39,656**
33	$312,138	$37,133	$7,960		**$38,307**
34	$322,914	$38,247	$8,080		**$37,005**
35	$334,064	$39,394	$8,200		**$35,747**
36	$345,601	$40,576	$8,320		**$34,532**
37	$357,537	$41,793	$8,440		**$33,359**
38	$369,888	$43,047	$8,560		**$32,226**
39	$382,666	$44,338	$8,680		**$31,132**
40	$395,888	$45,669	$140,800		**$38,890**
41	$409,568	$47,039	$8,920		**$29,054**
42	$423,722	$48,450	$9,040		**$28,068**
43	$438,367	$49,903	$9,160		**$27,116**
44	$453,520	$51,400	$9,280		**$26,197**
45	$469,199	$52,942	$9,400		**$25,309**
46	$485,422	$54,531	$9,520		**$24,451**
47	$502,207	$56,167	$9,640		**$23,622**
48	$519,575	$57,852	$9,760		**$22,822**
49	$537,546	$59,587	$9,880		**$22,049**
50	$556,140	$61,375	$10,000		**$21,303**
				Total	**$4,496,227**

Life Cycle Cost for FCU/DOAS w/Ice Storage

Year	Utility Cost	Annual Maintenance Cost	Replacement Cost	Initial Cost	Present Worth
0				$1,630,881	**$1,630,881**
1	$87,939	$14,420			**$95,663**
2	$90,992	$14,853			**$92,448**
3	$94,150	$15,298			**$89,343**
4	$97,419	$15,757			**$86,342**
5	$100,801	$16,230			**$83,442**
6	$104,302	$16,717			**$80,640**
7	$107,923	$17,218			**$77,932**
8	$111,671	$17,735			**$75,316**
9	$115,550	$18,267			**$72,787**
10	$119,563	$18,815			**$70,344**
11	$123,716	$19,379			**$67,984**
12	$128,014	$19,961			**$65,703**
13	$132,461	$20,559			**$63,498**
14	$137,064	$21,176			**$61,368**
15	$141,826	$21,812	$8,700		**$62,463**
16	$146,754	$22,466	$8,880		**$60,329**
17	$151,854	$23,140	$9,060		**$58,267**
18	$157,131	$23,834	$9,240		**$56,275**
19	$162,592	$24,549	$9,420		**$54,351**
20	$168,244	$25,286	$9,600		**$52,492**
21	$174,092	$26,044	$9,780		**$50,697**
22	$180,144	$26,825	$9,960		**$48,964**
23	$186,406	$27,630	$10,140		**$47,289**
24	$192,887	$28,459	$10,320		**$45,672**
25	$199,593	$29,313	$26,363		**$47,033**
26	$206,533	$30,192	$10,680		**$42,602**
27	$213,715	$31,098	$10,860		**$41,146**
28	$221,147	$32,031	$11,040		**$39,739**
29	$228,838	$32,992	$11,220		**$38,381**
30	$236,797	$33,982	$11,400		**$37,069**
31	$245,033	$35,001	$11,580		**$35,802**
32	$253,556	$36,051	$11,760		**$34,579**
33	$262,376	$37,133	$11,940		**$33,398**
34	$271,503	$38,247	$12,120		**$32,258**
35	$280,949	$39,394	$12,300		**$31,156**
36	$290,723	$40,576	$12,480		**$30,093**
37	$300,839	$41,793	$12,660		**$29,066**
38	$311,307	$43,047	$12,840		**$28,074**
39	$322,139	$44,338	$13,020		**$27,117**
40	$333,349	$45,669	$13,200		**$26,192**
41	$344,950	$47,039	$13,380		**$25,300**
42	$356,956	$48,450	$13,560		**$24,438**
43	$369,380	$49,903	$13,740		**$23,605**
44	$382,237	$51,400	$13,920		**$22,801**
45	$395,542	$52,942	$14,100		**$22,025**
46	$409,311	$54,531	$14,280		**$21,276**
47	$423,560	$56,167	$14,460		**$20,552**
48	$438,306	$57,852	$14,640		**$19,853**
49	$453,566	$59,587	$14,820		**$19,178**
50	$469,358	$61,375	$15,000		**$18,526**
				Total	**$4,021,749**

Life Cycle Cost for WSHP w/Sea Loop

Year	Utility Cost	Annual Maintenance Cost	Replacement Cost	Initial Cost	Present Worth
0				$1,800,482	**$1,800,482**
1	$67,929	$17,510			**$79,850**
2	$70,287	$18,035			**$77,144**
3	$72,727	$18,576			**$74,531**
4	$75,252	$19,134			**$72,006**
5	$77,864	$19,708			**$69,567**
6	$80,568	$20,299			**$67,212**
7	$83,365	$20,908			**$64,936**
8	$86,260	$21,535			**$62,738**
9	$89,256	$22,181			**$60,614**
10	$92,356	$22,847			**$58,563**
11	$95,564	$23,532			**$56,582**
12	$98,884	$24,238			**$54,667**
13	$102,319	$24,965			**$52,818**
14	$105,873	$25,714			**$51,032**
15	$109,552	$26,485			**$49,306**
16	$113,358	$27,280			**$47,639**
17	$117,298	$28,098			**$46,029**
18	$121,374	$28,941			**$44,473**
19	$125,592	$29,810			**$42,970**
20	$129,957	$30,704	$14,400		**$45,239**
21	$134,474	$31,625	$14,670		**$43,658**
22	$139,148	$32,574	$14,940		**$42,132**
23	$143,986	$33,551	$15,210		**$40,659**
24	$148,991	$34,557	$15,480		**$39,238**
25	$154,171	$35,594	$31,613		**$40,789**
26	$159,532	$36,662	$16,020		**$36,542**
27	$165,079	$37,762	$16,290		**$35,265**
28	$170,819	$38,895	$16,560		**$34,032**
29	$176,760	$40,062	$16,830		**$32,843**
30	$182,907	$41,263	$17,100		**$31,695**
31	$189,269	$42,501	$17,370		**$30,588**
32	$195,852	$43,776	$17,640		**$29,519**
33	$202,664	$45,090	$17,910		**$28,488**
34	$209,714	$46,442	$18,180		**$27,494**
35	$217,010	$47,836	$18,450		**$26,534**
36	$224,560	$49,271	$18,720		**$25,609**
37	$232,373	$50,749	$18,990		**$24,715**
38	$240,458	$52,271	$19,260		**$23,854**
39	$248,825	$53,839	$19,530		**$23,022**
40	$257,483	$55,455	$19,800		**$22,220**
41	$266,444	$57,118	$20,070		**$21,447**
42	$275,717	$58,832	$20,340		**$20,700**
43	$285,313	$60,597	$20,610		**$19,980**
44	$295,243	$62,415	$20,880		**$19,285**
45	$305,520	$64,287	$21,150		**$18,615**
46	$316,155	$66,216	$21,420		**$17,968**
47	$327,160	$68,202	$21,690		**$17,344**
48	$338,550	$70,248	$21,960		**$16,742**
49	$350,336	$72,356	$22,230		**$16,161**
50	$362,534	$74,526	$22,500		**$15,601**
				Total	**$3,801,138**

Appendix C: Life Cycle Cost Analysis

Life Cycle Cost			
Year	WSHP w/Sea Loop	VAV w/Ice Storage	FCU/DOAS w/Ice Storage
0	$1,800,482	$1,704,898	$1,630,881
1	$1,880,331	$1,817,075	$1,726,544
2	$1,957,475	$1,925,461	$1,818,992
3	$2,032,006	$2,030,185	$1,908,334
4	$2,104,012	$2,131,371	$1,994,676
5	$2,173,579	$2,229,139	$2,078,118
6	$2,240,791	$2,323,606	$2,158,757
7	$2,305,727	$2,414,883	$2,236,689
8	$2,368,465	$2,503,079	$2,312,005
9	$2,429,079	$2,588,299	$2,384,792
10	$2,487,643	$2,670,643	$2,455,137
11	$2,544,224	$2,750,208	$2,523,120
12	$2,598,892	$2,827,090	$2,588,823
13	$2,651,710	$2,901,379	$2,652,321
14	$2,702,742	$2,973,163	$2,713,689
15	$2,752,048	$3,042,528	$2,776,152
16	$2,799,687	$3,109,554	$2,836,481
17	$2,845,716	$3,174,321	$2,894,748
18	$2,890,189	$3,236,906	$2,951,023
19	$2,933,158	$3,297,382	$3,005,374
20	$2,978,397	$3,382,284	$3,057,866
21	$3,022,055	$3,440,330	$3,108,563
22	$3,064,188	$3,496,398	$3,157,527
23	$3,104,847	$3,550,557	$3,204,816
24	$3,144,085	$3,602,871	$3,250,489
25	$3,184,873	$3,656,327	$3,297,522
26	$3,221,416	$3,705,139	$3,340,124
27	$3,256,681	$3,752,289	$3,381,269
28	$3,290,713	$3,797,834	$3,421,008
29	$3,323,555	$3,841,829	$3,459,389
30	$3,355,250	$3,884,328	$3,496,458
31	$3,385,838	$3,925,380	$3,532,260
32	$3,415,357	$3,965,036	$3,566,839
33	$3,443,846	$4,003,343	$3,600,237
34	$3,471,339	$4,040,349	$3,632,495
35	$3,497,874	$4,076,096	$3,663,651
36	$3,523,482	$4,110,628	$3,693,744
37	$3,548,198	$4,143,987	$3,722,810
38	$3,572,051	$4,176,214	$3,750,885
39	$3,595,074	$4,207,345	$3,778,002
40	$3,617,294	$4,246,235	$3,804,194
41	$3,638,741	$4,275,290	$3,829,494
42	$3,659,441	$4,303,358	$3,853,931
43	$3,679,421	$4,330,474	$3,877,537
44	$3,698,706	$4,356,671	$3,900,338
45	$3,717,321	$4,381,980	$3,922,363
46	$3,735,289	$4,406,430	$3,943,639
47	$3,752,633	$4,430,053	$3,964,191
48	$3,769,375	$4,452,875	$3,984,044
49	$3,785,537	$4,474,924	$4,003,222
50	$3,801,138	$4,496,227	$4,021,749

Cost Comparison			
Year	WSHP vs VAV	WSHP vs FCU	VAV vs FCU
0	$95,583	$169,601	$74,017
1	$63,256	$153,788	$90,532
2	$32,014	$138,483	$106,469
3	$1,821	$123,671	$121,851
4	-$27,359	$109,336	$136,695
5	-$55,560	$95,462	$151,022
6	-$82,815	$82,034	$164,849
7	-$109,156	$69,038	$178,194
8	-$134,614	$56,460	$191,075
9	-$159,219	$44,287	$203,507
10	-$183,000	$32,506	$215,506
11	-$205,984	$21,104	$227,088
12	-$228,199	$10,069	$238,267
13	-$249,669	-$611	$249,058
14	-$270,422	-$10,948	$259,474
15	-$290,480	-$24,104	$266,375
16	-$309,866	-$36,794	$273,072
17	-$328,605	-$49,032	$279,573
18	-$346,717	-$60,834	$285,883
19	-$364,224	-$72,215	$292,008
20	-$403,886	-$79,469	$324,418
21	-$418,274	-$86,508	$331,766
22	-$432,211	-$93,340	$338,871
23	-$445,710	-$99,970	$345,741
24	-$458,787	-$106,404	$352,383
25	-$471,453	-$112,648	$358,805
26	-$483,723	-$118,708	$365,015
27	-$495,608	-$124,589	$371,020
28	-$507,122	-$130,296	$376,826
29	-$518,274	-$135,834	$382,441
30	-$529,077	-$141,207	$387,870
31	-$539,542	-$146,422	$393,120
32	-$549,679	-$151,482	$398,197
33	-$559,498	-$156,392	$403,106
34	-$569,009	-$161,156	$407,854
35	-$578,222	-$165,778	$412,445
36	-$587,146	-$170,262	$416,884
37	-$595,790	-$174,613	$421,177
38	-$604,162	-$178,833	$425,329
39	-$612,272	-$182,928	$429,344
40	-$628,941	-$186,900	$442,041
41	-$636,549	-$190,753	$445,796
42	-$643,917	-$194,491	$449,427
43	-$651,054	-$198,116	$452,938
44	-$657,965	-$201,632	$456,333
45	-$664,659	-$205,043	$459,616
46	-$671,141	-$208,350	$462,791
47	-$677,420	-$211,558	$465,861
48	-$683,499	-$214,669	$468,830
49	-$689,387	-$217,686	$471,702
50	-$695,089	-$220,611	$474,478

	System			WSHP w/Sea Loop		VAV w/Ice Storage		FCU/DOAS w/Ice Storage	
Utilities	Annual Increase	Rate	units	Annual Consumption	Cost	Annual Consumption	Cost	Annual Consumption	Cost
On-Peak Electricity Consumption (kWh)	3.5%	0.1614	$/kWh	253377	$40,895.10	388921	$62,771.92	341119	$55,056.66
On-Peak Electricity Demand (kW)	3.5%	9.75	$/KW	1272	$12,402.49	1906	$18,580.58	1643	$16,014.86
Off-Peak Electricity Consumption (kWh)	3.5%	0.085	$/kWh	149008	$12,665.64	206273	$17,533.17	168685	$14,338.21
City Water (cubic feet)	2.5%	0.02	$/ft³	62964	$1,259.27	215359	$4,307.19	81010	$1,620.21
City Sewer (gal)	2.5%	0.003	$/gal	235500	$706.50	805500	$2,416.50	303000	$909.00
Total Present Worth Cost/Year					**$67,929.00**		**$105,609.35**		**$87,938.95**

On-peak and off-peak electricity accounts for 5% supplied by photovoltaic cells and is deducted from total annual consumption. City sewer is estimated to be half of city water use.

Blue values indicate where the total present worth of the Water Source Heat Pump with Sea Loop system costs less than each of the other systems. At year 4, the WSHP system begins to cost less than the VAV system and at year 13 it begins to cost less than the FCU system.

Appendix C: Life Cycle Cost Analysis

FCU/DOAS w/Ice Storage

Item	Quantity		Material		Labor		Total Cost
	# Units	Unit	Unit Rate	Total	Unit Rate	Total	
Energy Recovery Ventilator with DX cooling and HGR	12000	CFM	$6	$72,000	$2	$24,000	$96,000
Fan Coil Units	50	EA	$2,000	$100,000	$1,000	$50,000	$150,000
HVAC Ductwork	46430	SF	$4	$185,720	$4	$185,720	$371,440
Hydronic Piping	46430	SF	$4	$185,720	$3	$139,290	$325,010
Water Cooled Chiller (100 tons)	1	EA	$45,000	$45,000	$15,000	$15,000	$60,000
Thermal Storage Sys (600 ton-hr)	1	EA	$100,000	$100,000	$75,000	$75,000	$175,000
Temperature Controls	46430	SF	$2	$69,645	$2	$92,860	$162,505
		Subtotal		$758,085		$581,870	$1,339,955
		Design Contingency	10%	$75,809	10%	$58,187	$133,996
		Sales Tax	7%	$5,307		$0	$5,307
		Subtotal		$839,200		$640,057	$1,479,257
		Overhead and Profit	5%	$41,960	5%	$32,003	$73,963
		City Multiplier	100%	$881,160	100%	$672,060	$1,553,220
		Total Contractor Cost		$881,160		$672,060	$1,553,220
		Construction Contingency	5%	$44,058	5%	$33,603	$77,661
		Total Cost		$925,218		$705,663	**$1,630,881**

WSHP w/Sea Loop

Item	Quantity		Material		Labor		Total Cost
	# Units	Unit	Unit Rate	Total	Unit Rate	Total	
Water Source Heat Pumps	165	Ton	$1,050	$174,000	$510	$84,000	$258,000
Energy Recovery Ventilator with DX cooling and HGR	12000	CFM	$6	$72,000	$2	$24,000	$96,000
HVAC Ductwork	46430	SF	$4	$185,720	$4	$185,720	$371,440
Hydronic Piping	46430	SF	$2	$92,860	$3	$139,290	$232,150
Geothermal Loopfield (300 foot deep = 1.5 tons)	120	EA	$1,000	$120,000	$2,000	$240,000	$360,000
Temperature Controls	46430	SF	$2	$69,645	$2	$92,860	$162,505
		Subtotal		$712,225		$767,870	$1,480,095
		Design Contingency	10%	$71,223	10%	$76,787	$148,010
		Sales Tax	7%	$4,986		$0	$4,986
		Subtotal		$788,433		$844,657	$1,633,090
		Overhead and Profit	5%	$39,422	5%	$42,233	$81,655
		City Multiplier	100%	$827,855	100%	$886,890	$1,714,745
		Total Contractor Cost		$827,855		$886,890	$1,714,745
		Construction Contingency	5%	$41,393	5%	$44,344	$85,737
		Total Cost		$869,247		$931,234	**$1,800,482**

VAV w/Ice Storage

Item	Quantity		Material		Labor		Total Cost
	# Units	Unit	Unit Rate	Total	Unit Rate	Total	
Central Air Handling Unit (low end)	56000	CFM	$3	$168,000	$2	$112,000	$280,000
VAV Units	50	EA	$1,600	$80,000	$800	$40,000	$120,000
HVAC Ductwork	46430	SF	$4	$185,720	$4	$185,720	$371,440
Hydronic Piping	46430	SF	$2	$92,860	$3	$139,290	$232,150
Water Cooled Chiller (100 tons)	1	EA	$45,000	$45,000	$15,000	$15,000	$60,000
Thermal Storage Sys (790 ton-hr)	1	EA	$100,000	$100,000	$75,000	$75,000	$175,000
Temperature Controls	46430	SF	$2	$69,645	$2	$92,860	$162,505
		Subtotal		$741,225		$659,870	$1,401,095
		Design Contingency	10%	$74,123	10%	$65,987	$140,110
		Sales Tax	7%	$5,189		$0	$5,189
		Subtotal		$820,536		$725,857	$1,546,393
		Overhead and Profit	5%	$41,027	5%	$36,293	$77,320
		City Multiplier	100%	$861,563	100%	$762,150	$1,623,713
		Total Contractor Cost		$861,563		$762,150	$1,623,713
		Construction Contingency	5%	$43,078	5%	$38,107	$81,186
		Total Cost		$904,641		$800,257	**$1,704,898**

Appendix D: Hand Calculations

D.1: From Section 3.1: Weather Data - *Dry climate calculation:*

$$P_{in} < 0.44(TF - 19.5)$$
$$P_{in}: Annual\ precipitation\ (inches)$$
$$TF: Annual\ mean\ temperature\ (fahrenheit)$$
$$3.6" < 0.44(80^{\circ}F - 19.5)$$
$$3.6" < 26.6"$$
$$\Rightarrow Dry\ (B)\ Climate$$

D.2: From Section 4.4: Creative High Performance Green Design - *Water fountain spray cooling tower calculation:*

Assuming $RH =$ 50%, $T_{db} =$ 39°C

Find rate of evaporation per area $W_{jet,pond}\left(\frac{kg}{sm^2}\right)$:

$$W_{jet} = \frac{(P_w - P_a)[0.089 + 0.0782(V_{tot})]}{Y}$$

Where: $P_w =$ Saturation vapor pressure at water temperature
$P_a =$ Saturation vapor pressure at room air dew point
$V_{tot} =$ Velocity accounting for water jet and wind
$Y =$ Latent heat of evaporation

$$P_w: Determined\ via\ weather\ data$$
$$\Rightarrow P_w = 6.993\,kPa$$

$$P_a: Determined\ via\ weather\ data$$
$$\Rightarrow P_a = 3.264\,kPa$$

$$V_{tot} = \sqrt{V_{avg}^2 + V_{wind}^2}$$
$$V_{wind}: Annual\ average\ via\ American\ Weather\ Service$$
$$\rightarrow V_{wind} = 4.10\frac{m}{s}$$

$$V_{avg}: P_1 + \frac{1}{2}\rho V_1^2 + \rho g y_1 = P_2 + \frac{1}{2}\rho V_2^2 + \rho g y_2$$
$$with:\ PE = 0\ @\ ground$$
$$and\ \ KE = 0\ @\ peak$$
$$and\ \ P_1 = P_2 = P_{atm}$$
$$and\ \ equal\ densities$$

$$\frac{1}{2}V_1^2 = g y_2$$
$$V_1 = \sqrt{2(g)(y_2)}$$
$$V_1 = \sqrt{2(9.8\frac{m}{s})(2m)}$$
$$\rightarrow V_1 = 6.26\frac{m}{s}$$
$$\rightarrow V_2 = 0\ (velocity\ at\ peak)$$
$$V_{avg} = \frac{V_1 + V_2}{2}$$
$$V_{avg} = \frac{6.26 + 0}{2}$$
$$\rightarrow V_{avg} = 3.13\frac{m}{s}$$

$$V_{tot} = \sqrt{3.13^2 + 5.17^2}$$
$$\Rightarrow V_{tot} = 6.04\frac{m}{s}$$

$$Y = Constant$$
$$\Rightarrow Y = 2260\frac{kJ}{kg}$$

$$W_{jet} = \frac{(6.993 - 4.10)[0.089 + 0.0782(6.04)]}{2260}$$
$$\Rightarrow W_{jet} = 0.000716\frac{kg}{sm^2}$$

$$W_{pond} = \frac{(P_w - P_a)[0.089 + 0.0782(V_{wind})]}{Y}$$
$$W_{pond} = \frac{(6.993 - 4.10)[0.089 + 0.0782(5.17)]}{2260}$$
$$\Rightarrow W_{pond} = 0.000631\frac{kg}{sm^2}$$

Find surface areas $SA_{jet,pond}\,(m^2)$:

$$SA_{jet} = 2\pi rh + \pi r^2$$
$$SA_{jet} = 2\pi(0.15)(2) + \pi(0.15)^2$$
$$SA_{jet} = 2.03m^2$$
$$SA_{jet,tot} = (8)(2.03)$$
$$SA_{jet,tot} = 16.2m^2$$

$$SA_{pond} = b * h$$
$$SA_{pond} = (10)(2)$$
$$SA_{pond} = 20m^2$$
$$SA_{pond,tot} = (2)(20)$$
$$SA_{pond,tot} = 40m^2$$

Find rate of evaporation $R_{evap}\left(\frac{kg}{s}\right)$:

$$R_{evap} = (W_{jet} * SA_{jet,tot})(W_{pond} * SA_{pond,tot})$$
$$(0.000716 * 16.2) + (0.000631 * 40)$$
$$\Rightarrow R_{evap} = 0.0368\frac{kg}{s}$$

Convert to tons of cooling:

$$\#\ tons\ =\ (R_{evap}\frac{kg}{sm^2})(Y\frac{kJ}{kg})(3412\frac{BTU}{kWh})(\frac{1}{12000\frac{BTU}{ton}})$$
$$(0.0368\frac{kg}{sm^2})(2260\frac{kJ}{kg})(3412\frac{BTU}{kWh})(\frac{1}{12000\frac{BTU}{ton}})$$
$$\Rightarrow 30.6\ tons$$

D.3: From Section 6.1: Modeling Software - *Curtain wall U-value calculations:*

$U_o = \frac{U_{cg}A_{cg} + U_{eg}A_{eg} + U_f A_f}{A_{pf}}$ [ASHRAE Fund. 09 C.15 Eq. 7 P. 15.4].

Where: $U_x =$ U value
$A_x =$ Area
$o =$ Overall
$cg =$ Center Glass
$eg =$ Edge Glass
$f =$ Frame

The example calculation for this process is as follows. This calculation represents the average U value across an entire two-window by four-window half-spandrel curtain wall (Designated U_{cw}) that is typical to the majority of the building:

$$U_{o,glass} = \frac{U_{cg}A_{cg} + U_{eg}A_{eg} + U_f A_f}{A_{pf}}$$
$$A_{cg} = \{[40 - 2(1.25in^2) - 2(2.5in^2)] * [36 - 2(1.25in^2) - 2(2.5in^2)]\}$$
$$A_{cg} = 925in^2$$
$$A_{eg} = \{[40 - 2(1.25in^2)] * [36 - 2(1.25in^2)] - A_{cg}\}$$
$$A_{cg} = 330in^2$$
$$A_f = (40 * 36) - \{[40 - 2(1.25in^2)] - [36 - 2(1.25in^2)]\}$$
$$A_f = 185in^2$$

$U_{cg} = 0.32$ $U_{eg} = 0.48$ $U_f = 1.63$ [ASHRAE Fund. 09 C15 T4 ID 21]

$$U_o = \frac{(0.32)(925) + (0.48 * 330) + (1.63)(185)}{(40)(36)}$$
$$U_o = 0.52$$
$$U_{o,spandrel} = \frac{U_{cg}A_{cg} + U_{eg}A_{eg} + U_f A_f}{A_{pf}}$$

U_{cg} is calculated to be 0.072, as it is an R-12 insulation-backed spandrel panel

U_{eg} is calculated as 40% between U_{cg} and U_{frame}, which remains unchanged (1.63):

$$U_{eg} = 0.072 + [0.40(1.63 - 0.072)]$$
$$U_{eg} = 0.70$$

Solving with the same areas used above:

$$U_{o,spandrel} = 0.42$$

Combining the four windows with the four spandrel panels and framing produces:

$$U_{cw} = \frac{[(n_{window})(U_{o,window})(A_{window})] + [(n_{spandrel})(U_{o,spandrel})(A_{spandrel})]}{(n_{total})(A_{window})}$$
$$U_{cw} = \frac{(4)(0.52)(40 * 36) + (4)(0.42)(40 * 36)}{(8)(40 * 36)}$$
$$\mathbf{U_{cw} = 0.47}$$

Appendix E: Noise Control Calculations

Element	63Hz	125Hz	250Hz	500Hz	1KHz	2KHz	4KHz	Comments
Custom Element	72	74	70	67	64	64	61	VFW vertical WHSP size 070
Straight Duct(RL)	-7	-9	-12	-22	-40	-40	-30	8ft rectangular lined duct
Elbow (In.sq.rct)	0	0	-1	-4	-7	-7	-7	lined square elbow with turning vanes, LOW VELOCITY 350FT/MIN
SubSum	**65**	**65**	**57**	**41**	**17**	**17**	**24**	
Elbow (regen.)	52	55	58	60	59	56	50	Regenerated sound from elbow.
SubSum	**65**	**65**	**61**	**60**	**59**	**56**	**50**	
Straight Duct(RL)	-12	-14	-19	-36	-40	-40	-40	13ft rectangular lined duct
Junction (X,atten.)	-8	-8	-8	-8	-8	-8	-8	X-junction , 2" inside corner radius
SubSum	**45**	**43**	**34**	**16**	**11**	**8**	**5**	
Junction (X,regen.)	49	50	50	50	49	47	43	Regenerated sound from junction.
SubSum	**50**	**51**	**50**	**50**	**49**	**47**	**43**	
Straight Duct(RL)	-33	-28	-24	-35	-40	-40	-40	8ft rectangular lined duct
Elbow (In.sq.rct)	0	0	0	-1	-4	-7	-7	lined square elbow, with turning vanes
SubSum	**17**	**23**	**26**	**14**	**5**	**5**	**5**	
Elbow (regen.)	35	40	43	46	48	47	44	Regenerated sound from elbow.
SubSum	**35**	**40**	**43**	**46**	**48**	**47**	**44**	
Straight Duct(RL)	-8	-7	-6	-9	-27	-29	-12	2ft rectangular lined duct
End Reflection	-25	-19	-14	-9	-4	-2	-1	End reflection loss
Sum	**5**	**14**	**23**	**28**	**17**	**16**	**31**	

NC 33 **RC 20(H)** **33 dBA**

Figure E1.1: NC-35 compliant path report for air supplied at 350ft/min
(source: Trane Acoustics Program 2012)

Element	63Hz	125Hz	250Hz	500Hz	1KHz	2KHz	4KHz	Comments
Custom Element	72	74	70	67	64	64	61	VFW vertical WHSP size 070
Straight Duct(RL)	-7	-8	-11	-21	-40	-40	-29	8ft rectangular lined duct
Elbow (In.sq.rct)	0	0	-1	-4	-7	-7	-7	lined square elbow with turning vanes, LOW VELOCITY 300FT/MIN
SubSum	**65**	**66**	**58**	**42**	**17**	**17**	**25**	
Elbow (regen.)	60	63	66	66	64	60	52	Regenerated sound from elbow.
SubSum	**66**	**68**	**67**	**66**	**64**	**60**	**52**	
Straight Duct(RL)	-11	-13	-18	-35	-40	-40	-40	13ft rectangular lined duct
Junction (X,atten.)	-8	-8	-8	-8	-8	-8	-8	X-junction branch, 2" inside corner radius
SubSum	**47**	**47**	**41**	**23**	**16**	**12**	**5**	
Junction (X,regen.)	48	48	48	46	44	41	37	Regenerated sound from junction.
SubSum	**51**	**51**	**49**	**46**	**44**	**41**	**37**	
Straight Duct(RL)	-29	-26	-23	-34	-40	-40	-40	8ft rectangular lined duct
Elbow (In.sq.rct)	0	0	0	-1	-4	-7	-7	lined square elbow, with turning vanes
SubSum	**22**	**25**	**26**	**11**	**5**	**5**	**5**	
Elbow (regen.)	37	41	44	46	45	42	37	Regenerated sound from elbow.
SubSum	**37**	**41**	**44**	**46**	**45**	**42**	**37**	
Straight Duct(RL)	-8	-7	-6	-9	-27	-29	-12	2ft rectangular lined duct
End Reflection	-25	-19	-14	-9	-4	-2	-1	End reflection loss
Sum	**5**	**15**	**24**	**28**	**14**	**11**	**24**	

NC 27 **RC 18(RH)** **28 dBA**

Figure E1.2: NC-30 compliance path report for air supplied at 300ft/min
(source: Trane Acoustics Program 2012)

Element	63Hz	125Hz	250Hz	500Hz	1KHz	2KHz	4KHz	Comments
Custom Element	85	74	71	70	78	80	76	Shop Noise
Wall or Floor	0	0	-1	0	0	-1	1	Shop: Source Room
Trans Loss Val	-27	-30	-35	-38	-39	-40	-40	6x8x18 in. dense concrete block; 36 lb/block, STC-48
Rec Rm Corr	-9	-9	-9	-9	-9	-9	-9	Corridor
Wall or Floor	-5	-5	-2	-1	0	0	-1	Corridor
Trans Loss Val	-11	-20	-30	-35	-39	-37	-39	3-5/8" metal studs; 5/8" gyp-board each side, STC-38
Rec Rm Corr	-4	-4	-3	-4	-5	-4	-5	Office: Receiver Room
Sum	**29**	**6**	**5**	**5**	**5**	**5**	**5**	

NC < 15 **RC 5(H)** **12 dBA**

Figure E2.1: Shop to office transmission loss path report
(source: Trane Acoustics Program 2012)